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POWER

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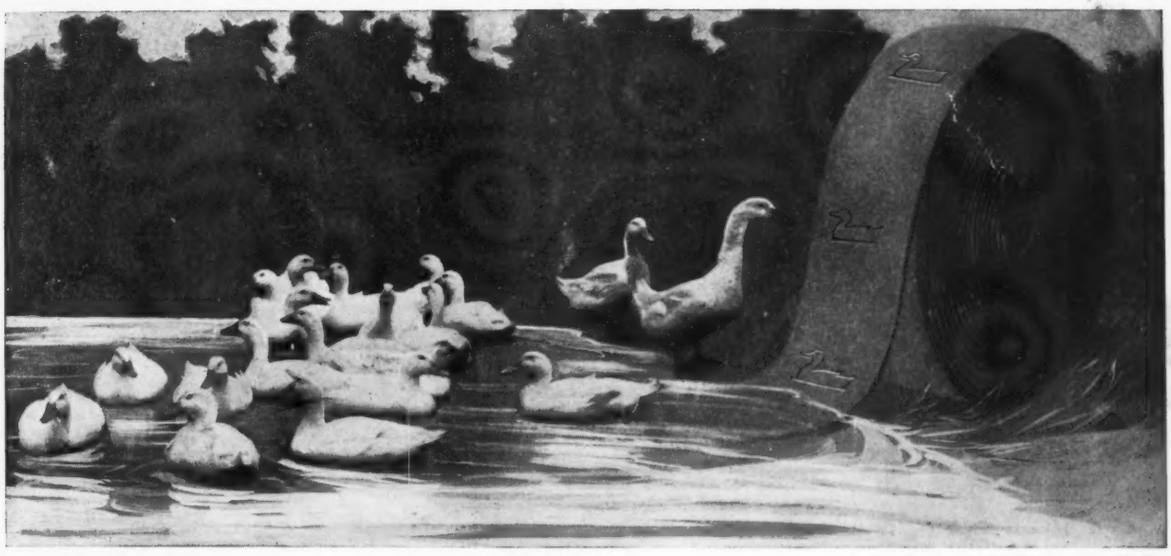

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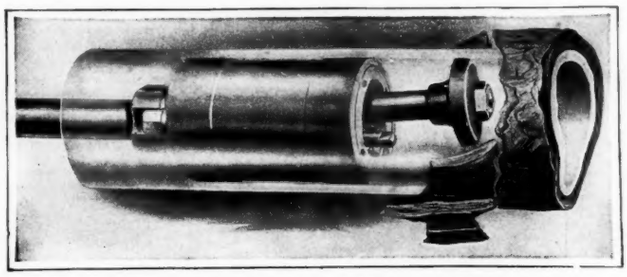

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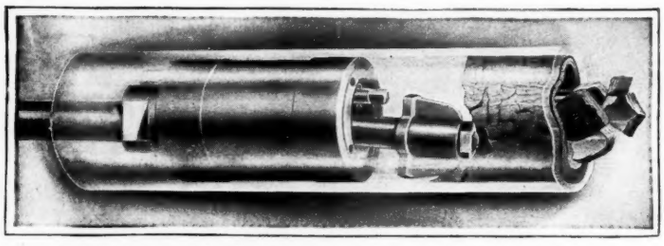

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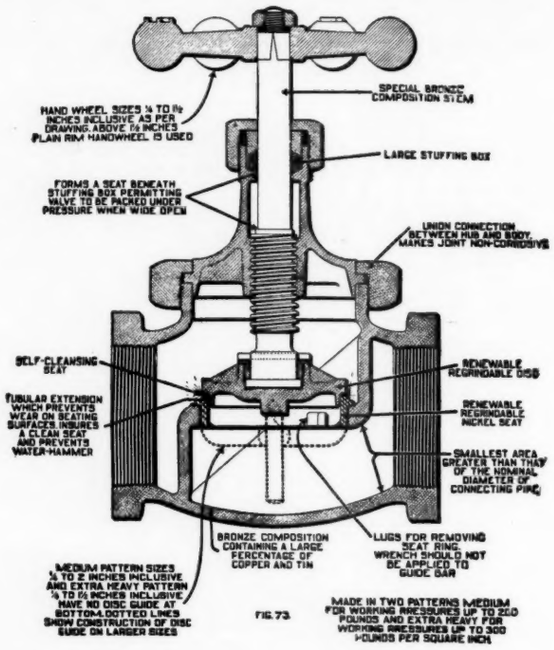
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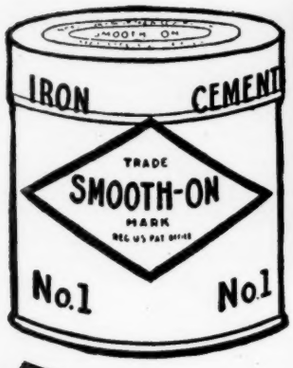

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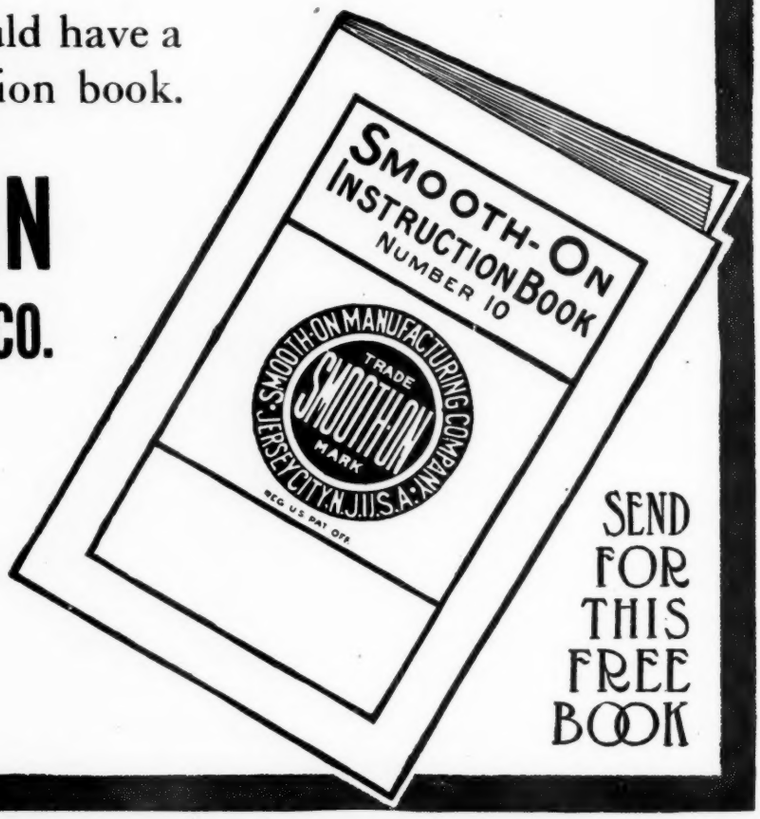

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POWER

Vol. 35 NEW YORK, APRIL 2, 1912 No. 14

"A MAN cannot turn anything over in his mind unless he knows it."

How strongly this applies to the power plant, where men must not only know a thing but know it well. The efficiency of the plant depends upon that of each individual in it, and individual efficiency is dependent upon his knowledge of his work. Where such work is potent with danger, and pregnant with liabilities to costly accidents, the skill required to perform it must necessarily be high.

Skill is applied knowledge, and the broader the knowledge of the work to be done, the more skilled is the worker.

There are quite a few engineers who do not give their subordinates full access to the means of acquiring more definite knowledge of their particular duties; done intentionally, this is nothing short of criminal; if not, it is gross negligence, and cannot be justified by any process of reasoning.

It is not enough that a switchboard operator merely know the switches and instruments on the board; he should be thoroughly acquainted with the numerous circuits connected with it. If it is a remote-control, high-tension board, the reasons are multiplied because the complications are greater.

Such operators and the dynamo and motor tenders should be supplied with blueprints showing in detail the circuits of the board, machines, elevators and whatever other apparatus the plant maintains.

The steam-fitters and plumber should also be supplied with charts hung where they will facilitate locating the valves controlling the lines of the steam and water systems.

Such charts or prints fail of their purpose when locked in the drawer of the chief's desk. One might as well have a telephone without a directory.

Engineers should have ready access to duplicates of these charts that they may familiarize themselves with the locations to which the charts refer. When trouble arises is no time to gather information as to the remedy for the trouble; the men must know on the instant what to do, and how best to do it. The uses of such means of information is highly necessary if the chief would have his men become better acquainted with the plant.

While the chief is to blame for such conditions, the engineers and those doing other work about the plant are oftentimes equally at fault. They are often reluctant to correctly and thoroughly answer the questions of the firemen, oilers, wipers, etc., and they fail to demonstrate to them where possible.

Engineering is greater than the individual engaged in it, and it is this mutual aid which will make for the betterment of the vocation.

Winding Engines of the Ashley Planes

By Warren O. Rogers

There are three planes at Ashley, Penn., over which the freight cars handled by the Central R. R. of New Jersey are drawn up the mountain.

Each plane is equipped with a twin-winding engine and drives two winding drums over which a 2½-in. cable runs.

The ends of each cable are attached to a barney-car which pushes the loaded freight cars up the planes.

When going into Wilkes-Barre, Penn., from the east on the Lehigh Valley R.R., the traveler will see, as he looks down the mountain when about 16 miles from the station, two sets of planes; a third is invisible behind the hill. These are known as the "Ashley Planes," and for years it has been the practice of the Central R.R. of New Jersey to haul freight cars up these three planes to the top of the mountain.

Powerful Engines Required

The first impression received is that powerful engines must be used in hauling the loaded freight cars up these steep grades. Curiosity overcame the writer's inclination to take things easy last Thanksgiving Day, and armed with a camera, a street car for Ashley where the planes terminate was boarded. A ride of about 20 minutes and the bottom of No. 3 plane was reached. No. 2 plane is about half-way up the mountain and has the heaviest grade of the three, and No. 1 plane lies at the top of the mountain.

The original machinery installed at these planes served its purpose for years, but in 1902 the winding machinery of the three planes was, to a large extent, rebuilt by the Bethlehem Steel Co. In place of the ordinary cast-iron winding gears, cut gears were substituted. Later on it was found that the increased traffic on the planes demanded higher speed, and it was gradually increased until it had been doubled, and still the amount of freight to be handled increased.

New Engines Installed

The excessive loads under which the engines were forced to labor soon caused them to begin to break down, and the same company which had made the repairs in 1902 was given a contract to build engines of sufficient size to handle the freight traffic going up the planes. These engines were made chiefly of steel frames and the bedplates were so designed that they would fit on the old foundations and between the old winding drums. A special design of valve-gear was also placed on the engines. The throttle valve is operated by means of steam admitted to the valve piston, and the reversing gear is operated in the same manner. These arrangements call for very slight exertion on the part of the operator when handling the engine.

Winding Engine

In Fig. 1 is shown a view of the latest design of winding engine, which was made by the Bethlehem Steel Co. On the engine shaft is mounted a spur gear 70 in. in diameter which meshes with two 20-ft. 10-in. geared sheave wheels, in which there are four grooves for the winding rope. The two-segment spur gearwheel has 36 cut teeth with a pitch of 6⅛ in. The winding gearwheels are of the built-up type, the rims, segments, hub and spoke being bolted together by 1¾-in. bolts. The pinion is geared to a 70-in. crankshaft and each of the two winding wheels is mounted on a separate 19-in. shaft.

The gear rims of both the pinion and the winding wheels are bolted to the body of each gearwheel and thus facilitate the making of repairs in case of a breakdown.

The engines are of the twin vertical type and have cylinders 48 in. in diameter by 40 in. stroke; the piston rods are each 6 in. in diameter. Steam is admitted to the steam chest of each cylinder through a 15-in. steam pipe and is exhausted to the atmosphere through an 18-in. exhaust pipe.

The engines at Nos. 2 and 3 planes are practically of the same size and design as the new unit, except that the latter, which was built in 1909, has a different valve-gear, as shown in detail in Fig. 2.

The cables used in hauling the cars up the planes are wound around the winding wheels four times, and pass out of the engine room under the floor to the center of the double track, sheave wheels being used where the cables make a bend.

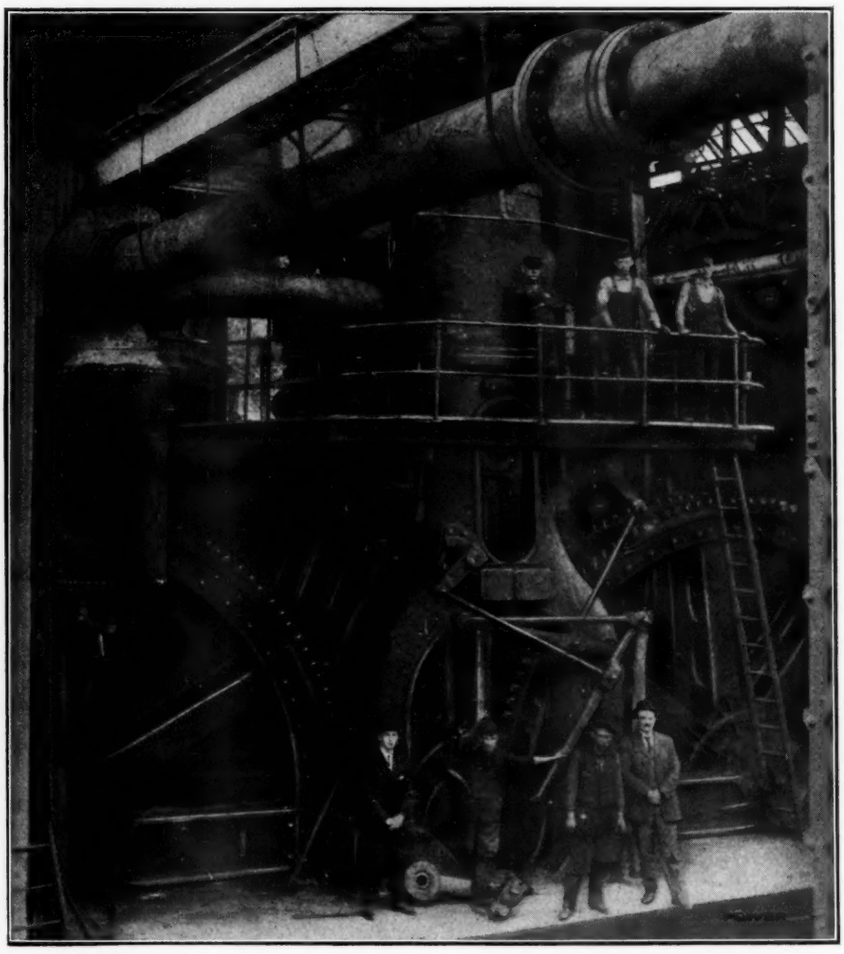


Fig. 1. The New Winding Engine at No. 1 Plane

No. 1 Plane

In Fig. 3 is shown, looking toward the bottom, a view of No. 1 plane. The barney car is on the down trip, and the method of taking care of the cable is also shown. In the pit between the rails there is one large sheave wheel about 4 ft. in diameter, and on either side of it there are small guide wheels about 8 in. in diameter.

The 2½-in. cable used on this plane is 4690 ft. in length, weighs 48,326 lb. and travels at a speed of 16 miles per hour. The time required to draw four 80-ton cars up No. 1 plane is 3½ min. The average tonnage of the rope is 10,000,000 tons. The ends of the haulage cable at each plane are secured to a barney car after the manner shown in Fig. 5.

Tail Rope

A 1½-in. tail rope is secured to the rear of each barney car having a tonnage twice that of the hauling cable, or 20,000,000. The tail rope runs on the same guide wheels that are used by the haulage rope. The slack is taken up by a compensating device, shown in Fig. 6. This consists of a sheave wheel mounted on a truck that is free to travel on a short incline track. The truck is weighted and as slack occurs in the tail cable from expansion or stretch, it is taken up by the compensating device.

No. 2 Plane

A view of No. 2 plane from the top is shown in Fig. 4. This plane has a grade of 14 per cent., and it requires 3 min. to draw three 80-ton gondola cars to the top at a speed of 20 miles an hour. The cable is 2¾ in. in diameter, is 3670 ft. long and weighs 44,512 lb. The hauling rope has a useful capacity of 630,000 tons

No. 3 Plane

The bottom, No. 3, plane has a grade of 5 per cent. and is equipped with a 2½-in. steel cable 5780 ft. in length. It weighs 59,792 lb. and runs at a speed of 13 miles per hour. The load on the plane is six 80-ton cars and is handled in 4½ min. Fig. 5 shows a train starting from the foot of No. 3 plane. The manner of handling the cars by means of a barney car is shown in Fig. 7.

The old barney cars were made of cast iron, but a new steel car has recently been built and a new motion of changing the gage of the wheels has been adopted. A new cable fastening has also been adopted, which makes it impossible for the rope to pull out of its fastening.

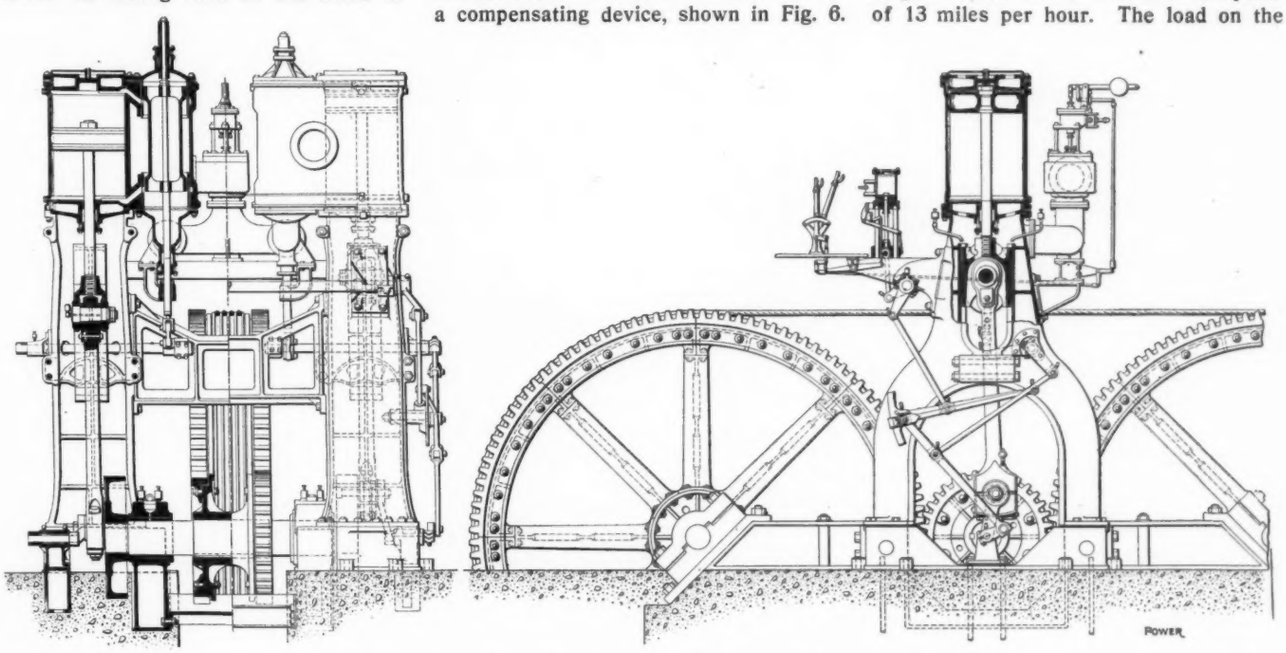


Fig. 2. Front and Side Elevation of Winding Engine at No. 1 Plane

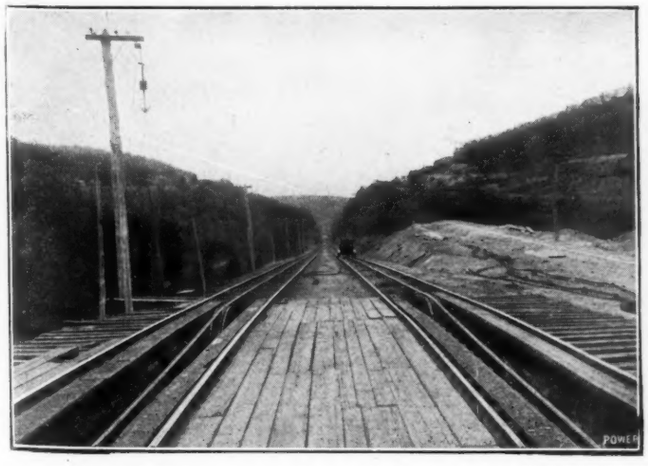


Fig. 3. Looking Down No. 1 Plane

Fig. 4. Looking Down No. 2 Plane

Accidents Not Frequent

It is not often that an accident occurs, but when the cars do get away a genuine wreck follows. In Fig. 8 is

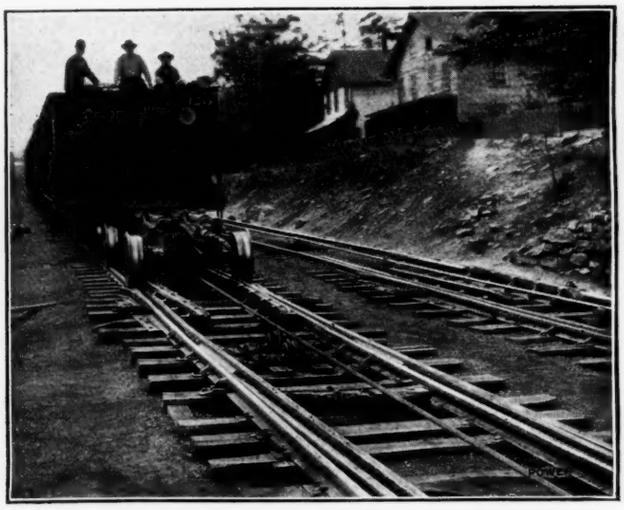


FIG. 5. BARNEY CAR ON NO. 1 PLANE; ALSO SWITCH ARRANGEMENT

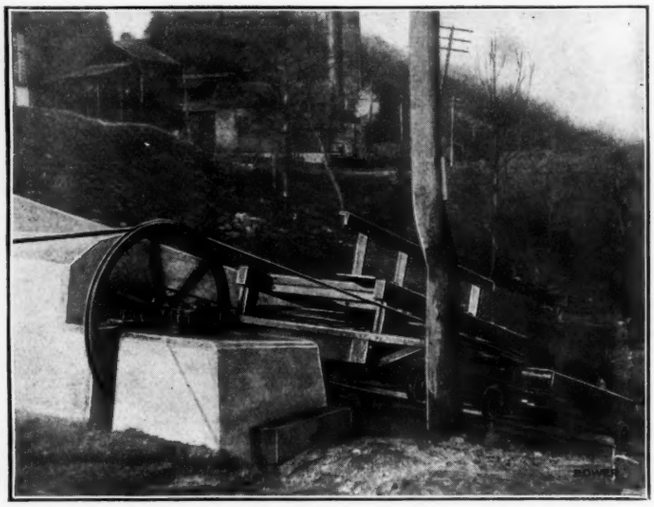


FIG. 6. HOW SLACK IN THE TALE ROPE IS TAKEN UP

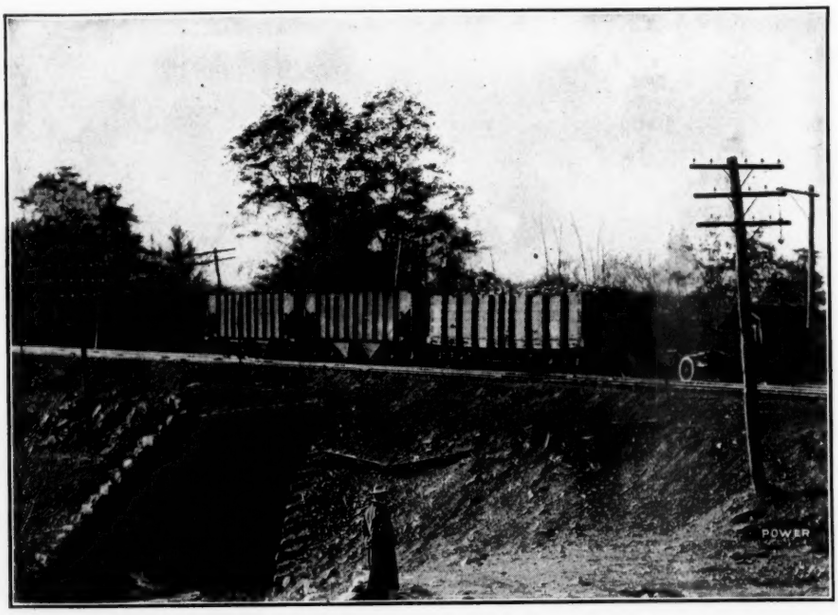


FIG. 7. BARNEY CAR AND LOADED COAL CARS ON NO. 2 PLANE

shown a spill which resulted from the cable pulling out of its fastenings on the barney car when the train of four loaded gondola coal cars was about halfway up No. 1 plane. Although the cars were wrecked, fortunately no one was hurt, owing, it is said, to the sprinting ability of the men who were employed at the foot of the plane.

AUTOMATIC SWITCHES

An examination of Fig. 5 shows an ingenious arrangement for automatically switching the barney car into the pit at the bottom of the plane. It will be noticed that, although the regular freight-car trucks are made with the wheel flanges on the inside of the rail, the barney car is made with the flanges on the outside of the rail.

Referring to the track construction in Fig. 5, it will be seen that there are three sets of rails on the near side of the barney car. The outside set is for the regular freight cars, but the four inside rails are for the barney car. In the illustration the barney car has just taken the wide-gage rails. If it were returning to the pit from which it has just emerged, the wheels, upon striking the turned-out end of the rails, would be forced closer to the body of the car and would travel on the inner rails, with the outside of the flange bearing against the inside of the outer rails.

When the wheels of the barney car strikes the point of the inside rails, the flanges strike the inside rails and are further forced toward the body of the car, taking the inside rails with the flanges bearing against the center rails. When the bottom of the pit is reached, the car runs through a spring switch onto the center rails, which rise abruptly to the track surface at the far end of the pit and run parallel with the outer rails until the frog is reached.

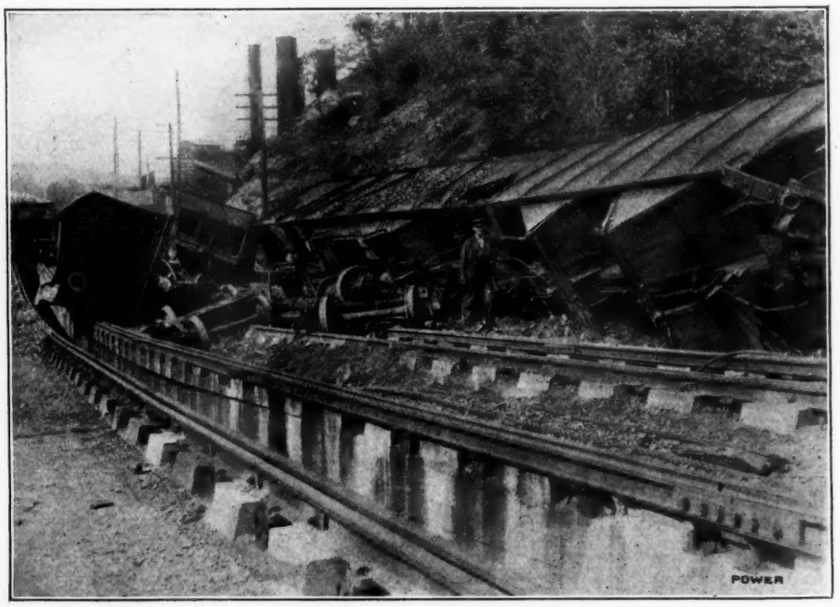


FIG. 8. RESULT OF A RUNAWAY

On the return trip, the car runs on the inner rails, but as it nears the point where the rails is turned out on the outside, the flanges of the barney-car wheels forces the rails out again and permits the tracks to assume their widest gage. As soon as the wheels have passed the points, the springs, which are in tension against the rails, force them back into place. The central rails are held in place by means of springs which force them against the outer rails. The barney car is made with four separate axles, each of which slides in a long square box, in the inner end of which is a spring which presses against the end of the axle.

An inner guide rail assists in forcing the outer rails apart as the barney-car wheel strikes them and the rails at the same time. Should the spring at the end of the axle break, the wheels would take the proper rails at the right place. The ordinary car trucks always take the outer rail.

Scale Prevention

By John Creen

For the treatment of moderate quantities of feed water in which carbonate of lime predominates, handy and effective apparatus may be contrived by the engineer out of one or more boiler tanks, in which the water is heated by exhaust steam through a system of pipe coils, soda or other chemicals being used as reagents. By this means very bad water may be purified of a large portion of its mineral matter, which is left behind in the tank. Feed-water heaters, whether using exhaust or live steam, answer the same purpose. In the lime process, 28 grains of caustic lime will give a total precipitate of 100 grains of carbonate, the 28 grains of caustic being themselves converted into 50 grains of carbonate.

But for various reasons, steam users often prefer to treat the water inside the boilers, in which case the use of caustic lime forms a too bulky precipitate, and some other reagent must therefore be employed. By using caustic soda, both carbonate and sulphate of lime can be thrown down, and the process is more rapid than with caustic lime and carbonate of soda. Suppose, for instance, a water contains 6 grains of carbonate, and 8.16 grains of sulphate of lime; by adding 3.75 grains of caustic soda to the water, these at once become 6.36 grains of carbonate of soda, and 6 grains of carbonate of lime are precipitated. The 6.36 grains of carbonate of soda decompose the 8.16 grains of sulphate of lime, and are converted into 8.52 grains of sulphate of soda, and another 6 grains of carbonate of lime are precipitated. Altogether, 12 grains of lime are thrown down, and the soda acts twice, first by absorbing an acid, and second by changing this acid for another. When used in this way, 31 grains of caustic soda will precipitate 100 grains of carbonate of lime. Soda therefore will remove nearly double the quantity removed by caustic lime from waters containing both carbonate and sulphate of lime, but the cost of treatment by soda is considerably more than by the lime treatment. The following formulas will be found useful:

1. Water in which carbonate of lime only is present—For treating 1000 gal. of water, 1060 grains of carbonate of soda, made caustic with 560 grains of burnt lime, are required for each grain of carbonate of lime per gallon. To produce the same result with soda, 1½ oz. of caustic soda for each grain of carbonate of lime present per gallon will be necessary for the treatment of 1000 gal.

2. Waters containing sulphate of lime only—For each grain of sulphate of lime per gallon, 1¾ oz. of pure carbonate of soda will be required to treat 1000 gal. of water.

3. When both carbonate and sulphate of lime are present—As explained for carbonate waters (1), 1½ oz. of caustic soda for each grain of carbonate of lime will precipitate the lime in 1000 gal., but this quantity will also throw down 1.36 grains of sulphate of lime at the same time, so that if the proportion of sulphate does not exceed the carbonate beyond the ratio of 8 to 6, the sulphate may be disregarded. If, however, sulphate of lime is present in a greater ratio, carbonate of soda should also be added. An excess of soda should be used to insure a proper reaction, which is greatly assisted by heating the water.

Unless the soda used is of the best quality and free from adulteration, leakages at the seams may be caused, or the plates and fittings may be attacked; the use of impure caustic soda has been the cause of much trouble in this direction.

When, however, scale has formed on the internal heating surfaces and shell of the boiler, there is an effective method (which the writer has practiced with success for many years) of removing it. Caustic potash answers this purpose, especially when the scale consists chiefly of mud, with sulphate and carbonate of lime and magnesia combined. Caustic potash is the concentrated lye sold in grocery stores, but if it is required in large quantities it should be purchased from wholesale druggists. Dissolve it in a barrel of water, say 40 lb. to the barrel, and then pump it into the boiler; this is about ⅛ lb. of potash per pound of water and is strong enough for the purpose.

The method employed by the writer was as follows: One boiler was cut out of service at noon on Saturday, the damper was nearly closed and the pressure let drop to 30 or 40 lb. The potash being ready, the mixture was pumped from a suitable tank into the boiler; continuing to pump clean water until the glass was full. The fire was then allowed to die out, and the boiler left in this condition for at least 24 hours, when the furnace was again lighted, and the steam pressure raised to 15 or 20 lb. The fire was then drawn and the boiler blown down; the bottom doors were taken off, all the chunks of scale which could not go through the blowoff cocks were removed, and the boiler was thoroughly washed out with water from a hose at high pressure. The boiler was then closed, filled with clean water and steam raised, and put into commission again. A repetition of this at every week-end for a few weeks will thoroughly clean the dirtiest boiler.

In handling caustic potash the utmost care must be used; it is truly caustic, and if a small portion splashes in one's face or eyes it will cause serious trouble. The same is true of the hands. For a very hard scale an acid purger should be used, as lime is alkaline and its antidote is an acid. Probably someone may exclaim loudly against using an acid purger, arguing that it will destroy the boiler. It is just here that trouble is likely to ensue in the hands of an inexperienced person. In the first place, an acid purger is largely diluted with water, and it attacks the scale first, having the greatest affinity for it. After that it would attack the boiler plates if allowed to remain in the boiler, but there can be no after effects of this character from a boiler purger, because it is no longer in the boiler when the scale is removed, the boiler cleaned out by hand, washed with a hose, filled up and blown out again before steam is raised. Of the two evils, it is best to choose the least, and in this case the use of a strong boiler purge is less than the injury and loss of fuel caused by scale.

Revenue in Public Water Power

A report of the new Conservation Commission recently submitted to the New York legislature finds that the waters of the state furnish a total flow having a capacity of 1,271,000 hp.

Ordinarily, development for manufacturing purposes is based upon an estimate of only 60 per cent. of the time of continuous flow. A well considered plan of conservation would yield an available horsepower of about 1,780,000. In an investigation before the foreign affairs committee of the House of Representatives it was shown that the Ontario Power Co., of Niagara, sells electric power for $9.40 per horsepower-year. At a rate of $30 for comparatively small developments it will be at once evident how large a source of income will be available to the state, whenever it develops its water powers.—*Scientific American.*

Cost of Isolated Plant Power

From a paper prepared by the American Engine Co., showing the cost of power as generated by noncondensing engines, the exhaust of which is used for heating during the winter season.

The method of arriving at the costs as herein outlined is applicable to any isolated plant.

Where steam heating is required, electric power may be generated as a by-product of the heating plant, at a cost of less than 2c. per kilowatt-hour in the majority of cases. The reasons for this low cost of generation may be summarized as follows:

First. The fixed costs, which go to make up a large share of the cost of power, are chargeable only against the additional investment for the plant by which electric power is generated, over and above the investment for the low-pressure steam-heating plant.

Second. The coal cost chargeable against electric power is reduced, during those periods when the exhaust steam is used for heating, to a small percentage of the coal consumed, represented by that part of the heat in the steam, which is lost between the high-pressure boiler and the heating system.

Third. The labor charge comprises only the additional labor required, as compared to a low-pressure heating plant. Furthermore, by reason of the additional labor, which is of a higher grade, greater economy is obtained in fuel consumption.

Fourth. While the charge for oil, waste, repairs and miscellaneous items is incurred entirely through the installation of the engine and generator, they amount to but a small proportion of the total expense.

Every isolated plant with exhaust-steam heating must be considered as a separate problem, because of the wide variation in load, cost of coal, water, labor, etc. The following case, which may be considered typical, will serve to outline the calculation whereby the cost of power may be determined, and secondly to indicate the influence on the cost per kilowatt-hour of the three most important factors, the fixed charges, the price of coal and the amount of exhaust steam used for heating.

Fixed Charges

An industrial plant operating 3000 hr. per year at an average load of 200 kw. will be assumed with two 100-kw. and one 50-kw. engine-driven dynamos installed. These, with suitable piping and auxiliaries, may be set down at $11,000.

Two 200-hp. boilers would be required. However, the total cost of these boilers would not be chargeable against power since boilers would also be required for the low-pressure heating plant. The cost per horsepower of low-pressure boilers, grates, stack, etc., is nearly the same as a similar high-pressure equipment. But as heat is lost and consumed between the high-pressure boiler and the heating system, a somewhat greater boiler capacity must be installed when power is to be generated. As the exhaust from an engine contains 80 to 90 per cent. of the heat given it, in the boiler, an increase in boiler capacity of 25 per cent. over the low-pressure equipment will easily take care of this item. Considering $18.75 per horsepower, including boiler, grates, stack setting, piping and labor of installation, the additional boiler cost would be

$$400\ hp. \times 20\% \times \$18.75 = \$1500$$

The two largest items of additional first cost, totaling $12,500, have been considered. It might, of course, be profitable to install other apparatus in a plant of this character, as, for instance, a feed-water heater. But as that apparatus should pay for itself out of the coal saved, it will not be considered under the items of additional first cost.

Of the total additional cost of $12,500, a certain fixed percentage must be charged off every year against the cost of power. The fixed charges would be:

	Per cent.
Interest at	5
Depreciation at	5
Insurance, real estate, upkeep, obsolescence	5
Total	15

And

$$0.15 \times \$12,500 = \$1875 \qquad (1)$$

the fixed charge per year against the additional investment for apparatus to generate power as a byproduct of the heating system.

Labor, Oil, Waste, Etc.

The cost for additional labor may be taken as one man at $800 per year, while repairs, oil waste, packing, etc., may be set down as $250, making a total of

$$\$1050 \qquad (2)$$

Coal

The coal chargeable against power comprises the coal equivalent of the heat lost between the high-pressure boiler and the heating system when the heating season is on; and, when no heating is being done, all of the coal burned is chargeable against power.

The cost for coal will depend therefore upon (a) the steam consumption of the engines; (b) the evaporation of the boiler; (c) the cost of coal, and (d) the percentage of the total amount of steam generated which is used by the heating system.

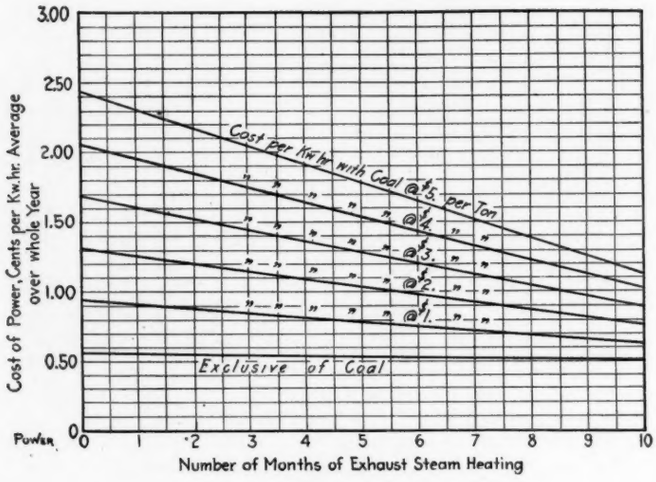

Cost of Power, Cents per Kw.hr. Average over whole Year
Cost per Kw.hr with Coal @ $5. per Ton
" " " " " @ 4. " "
" " " " " @ 3. " "
" " " " " @ 2. " "
" " " " " @ 1. " "
Exclusive of Coal
Power
Number of Months of Exhaust Steam Heating

Fig. 1. Total Cost

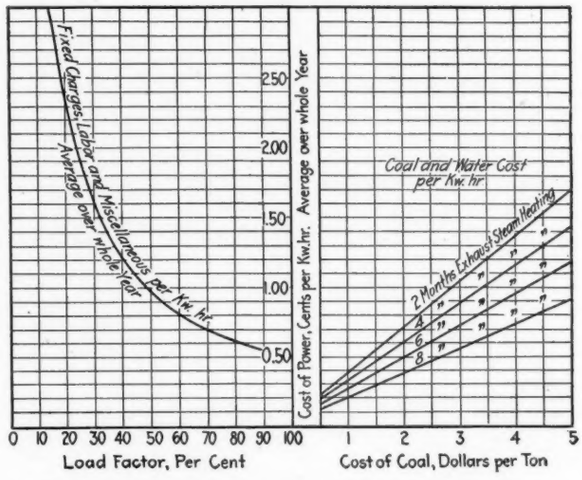

Fixed Charges, Labor and Miscellaneous per Kw.hr. Average over whole Year
Cost of Power, Cents per Kw.hr. Average over whole Year
Coal and Water Cost per Kw.hr
2 Months Exhaust Steam Heating
4 " " " "
6 " " " "
8 " " " "
Load Factor, Per Cent
Cost of Coal, Dollars per Ton

Fig. 2. Cost as Dependent upon Load Factor

(a) The steam consumption of a good high-speed engine for isolated-plant work is 26 lb. per horsepower-hour, running noncondensing, or 40 lb. per kilowatt-hour. To be on the safe side and to take care of the effect of reduced and fluctuating loads on the steam consumption, assume 60 lb. of steam per kilowatt-hour.

(b) Assume 8 lb. of steam per pound of coal.

(c) and (d) The cost of coal and the proportion of the total steam generated which is used in the heating system, are both variable. In the figures given herewith, the cost of coal ranges from $1 to $5 per ton. The amount of steam used in the heating system is taken care of by considering it proportional to the number of months that the exhaust steam is used for heating. During those months it is assumed that all the exhaust steam is used by the heating system. Strictly speaking, at the beginning and end of the heating season, there are periods when only a moderate amount of heating is required and only a portion of the exhaust steam is utilized. This decrease in heating load may be taken care of by assuming a shorter heating season. For example, suppose that during the four winter months all the exhaust steam is used in heating and that for 1½ months before and after this season the amount of steam used in the heating system varied from zero to full capacity. It would then be quite accurate to assume that the heating season of seven months would be equivalent to a season of 5½ months in which the heating load was always equivalent to the full exhaust of the engine.

Water

An average price for water is 10c. per thousand gallons. During the heating season it is assumed that only 30 per cent. of the water is returned to the boiler; in other periods all the water is lost.

With the fixed costs and labor, as already given, and calculating the coal and water charges for given conditions, the charts shown were obtained. The following example will serve to explain in detail the method of obtaining these figures:

Example

Assume the heating season equivalent to six months of full-load steam heating, when all the exhaust is used. During this period the coal chargeable against power is the equivalent of the heat lost between the high-pressure boiler and the heating system. Assume that 20 per cent. of the heat of the steam is lost in this manner. The other factors entering into the coal cost have already been given, i.e., load 200 kw.; steam consumption 60 lb. per kilowatt-hour; evaporation 8 lb. of steam per pound of coal; hours per year, 3000. With coal at $4, the cost for coal chargeable to power during the six months' heating period would be:

$$200 \times \frac{3000}{2} \times \frac{60}{8} \times 0.20 \times \frac{\$4}{2000} = \$900$$

The part of the year when there is no exhaust-steam heating, and those times when only part of the exhaust steam is used for heating, are equivalent to six months of straight noncondensing operation. The cost of coal is therefore

$$200 \times \frac{3000}{2} \times \frac{60}{8} \times \frac{\$4}{2000} = \$4500$$

To this coal cost must be added the cost for water, which may be set down at 10c. per thousand gallons. During the heating season, the heating system returns 80 per cent. of the boiler feed. The cost for water is then

$$200 \times \frac{3000}{2} \times \frac{60}{8.33} \times 0.20 \times \frac{10}{1000} = \$43.20$$

For the other six months the cost for water is as follows:

$$200 \times \frac{3000}{2} \times \frac{60}{8.33} \times \frac{10}{1000} = \$216$$

Therefore the total charges for generating power at the rate of 200 kw. for 3000 hr. a year are as follows:

Fixed charge	$1875
Labor, oil and miscellaneous	1050
Coal	4500
	900
Water	216
	43
Total	$8584

During the year there are developed 3000 × 200 = 600,000 kw.-hr., giving a cost per kw.-hr. of

$$\$8584 \div 500{,}000 = 1.431 \text{ cents}$$

In the same manner it is found that when the heating season is equivalent to only two months of full-load heating, the cost per kilowatt-hour is 1.85c. For the same conditions when the price of coal is $5 instead of $4, the cost per kilowatt-hour is 2.175c.

From these data the charts shown in Figs. 1 and 2 were prepared. In Fig. 2 the cost, exclusive of coal and water, is considered separately; being plotted against the load factor.

A load of 200 kw. for 3000 hr. per year is considered as a 100 per cent. load factor (although the plant capacity is 250 kw.). At this load the fixed charge, labor charge, etc., per kilowatt-hour is

$$\frac{\$1875 + \$1050}{600{,}000} = 0.488 \text{ cent}$$

Similarly, at 50 per cent. load factor the cost per kilowatt-hour, exclusive of coal and water, is

$$\frac{\$1875 + \$1050}{300{,}000} = 0.976 \text{ cent}$$

By similar calculations the curve is plotted, which may be used in connection with the curves at the right in order to study the cost of power under various conditions.

Suppose, for instance, that the load factor is 50 per cent., then the fixed charges, labor, etc., amount to $0.98. If the coal cost is $4 a ton and the number of months of exhaust-steam heating is 4; then the coal and water cost per kilowatt-hour is 1.17c. and the total cost of power per kilowatt-hour is 2.15c.

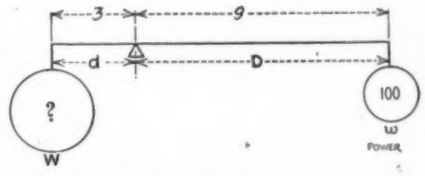
Study Questions

In accordance with the suggestion of a subscriber, who would like to work out some problems and be assured of the correctness of his results, but who finds the answers in the usual catechism too handy, we submit the following, the answers to which will be published next week:

1. A generator unit uses 16.5 lb. of steam per kilowatt-hour. What is the consumption per indicated horsepower-hour if the overall efficiency of the unit is 90 per cent.?

2. A boiler evaporates 8.5 lb. of water per pound of coal from feed having a temperature of 160 deg. into dry saturated steam of 135 lb. absolute pressure. What is the equivalent evaporation from and at 212 deg.?

3. What weight will be required at *W*



to balance a weight of 100 lb. at *w* when the distances are as given in the sketch?

4. How far would a body starting from rest drop in one second; in four seconds? Explain why, air resistance neglected.

5. If a boiler had its safety valve set to blow at 100 lb. gage at the sea level, what pressure would the gage show when the valve blew if the boiler were moved to an elevation where the pressure was one pound less per square inch?

Writing on electrical power in the Transvaal in *Cassier's Magazine* for February, Rowland Gascoyne gives the following schedule of prices as fixed by the Victoria Falls Power Co. for the supply of electrical energy.

Load factor, percentage	Price per unit (kw.-hr.) from Oct. 1, 1910, to Oct. 1, 1912, Pence	Price per unit after Sept. 30, 1912, Pence
10	2.512	2.475
15	1.754	1.717
20	1.375	1.338
30	0.995	0.958
40	0.807	0.770
50	0.692	0.655
60	0.617	0.580
70 and over	0.5617	0.525

The magnitude of this company's operations is understood when it is known that its capitalization is $30,000,000; its license allows an annual output up to 500 million kilowatt-hours, and its transmission lines extend over 50 miles.

Low Pressure Turbine Installation

By B. R. T. Collins

To meet the increased demand for power, a 1500-kw. low-pressure turbo-generator was installed in connection with one of the noncondensing Corliss engines.

According to the figures, the saving effected by this arrangement for the present year will amount to about twice the fixed charges on the additional investment.

A further demonstration of the economy to be obtained by the installation of a low-pressure turbine in connection with a Corliss engine has recently been made in the power plant of the Goodyear Tire & Rubber Co., of Akron, Ohio. A large proportion of the power developed by this plant is required to drive machinery in a department of the factory adjoining the engine room and is delivered to several lines of heavy shafting, with couplings direct to the machines, by a rope-drive system from a Corliss engine. The balance of the power required in the factory buildings, which cover an area of about 25 acres, is furnished by electric drive from a low-pressure turbo-generator using the exhaust steam from the engine together with that from various plant auxiliaries.

The Corliss engine is a 34 and 60 by 60-in. Hamilton horizontal cross-compound noncondensing machine rated at 2500 hp. and running at 80 r.p.m. It will be noted that a cylinder ratio of 1 to 3.1 is used in order to have an engine of low steam consumption when running noncondensing. The American system of rope drive is used to transmit the power from the engine to the line shafting.

There is also a 26 and 52 by 54-in. Fulton horizontal cross-compound condensing engine rated at 1200 hp. which is available for operation with the Hamilton engine and which also serves as a partial relay to it.

The low-pressure turbine is a 1500-kw. Allis-Chalmers machine direct connected to a three-phase, 60-cycle, 480-volt generator running at 1800 r.p.m.

The specified steam pressure at the low-pressure throttle is 15 lb. per square inch absolute or about 0.3 lb. gage at average atmospheric pressure. A back-pressure valve is placed in the exhaust line from the engine to the turbine and is adjusted to maintain a constant back pressure on the engine of about 1 lb. gage. This gives the engine a constant exhaust pressure to work against and at the same time prevents the vacuum from backing up through the turbine into the exhaust header which is 48 in. in diameter and 50 ft. in length and is more easily kept tight under pressure than under vacuum.

A high-pressure throttle is provided for supplying live steam to make up for any deficiency in the exhaust steam. The use of live steam is not likely to occur to any great extent, however, as the engine ordinarily operates 23 hr. per day and during the remaining two half-hour periods when it is shut down, the turbine load decreases also; moreover, all the auxiliaries which furnish a considerable percentage of the total exhaust steam to the turbine are in service 24 hr. per day.

The turbine condensing equipment consists of an Allis-Chalmers condenser, provided with two single-stage centrifugal pumps mounted on a common shaft coupled direct to a 175-hp. Terry steam turbine. One of these pumps, which has a double impeller, removes the condensed steam and water from the condenser body and the other delivers water from the cold well to the twin air ejectors. The water used by the latter pump is returned to the cold well and is used over again as its temperature is not increased materially by its passage through the ejector nozzles. A 28-in. vacuum is guaranteed with a full load of 1500 kw., 15 lb. absolute steam pressure at the throttle and injection water at not more than 70 deg. F.

As a partial relay for the low-pressure turbine, there is a 500-kw. Crocker-Wheeler motor-generator set, used ordinarily to convert alternating into direct current for the variable-speed motors. This can be operated inverted from a Cooper condensing engine direct-connected to a 300-kw. Westinghouse direct-current generator and thus provide relay capacity up to the overload limit of the engine-driven unit. This arrangement is also convenient for carrying the load on Sundays and at other times when the demand for power is light.

Later, it is proposed to provide more relay capacity by installing an auxiliary turbine which can be coupled to the motor-generator set in an emergency and which will provide sufficient power to generate 1000 kw. divided equally between alternating and direct current.

A test recently made on the low-pressure turbine showed a steam consumption of 32.8 lb. per kilowatt-hour output of the turbine alone at a steady load of 1292.7 kw. under the conditions specified which is 6.3 per cent. better than the maker's guarantee at this load.

Considering the engine and turbine as a unit, a water rate of 16.95 lb. per kilowatt-hour was obtained.

In order to make a comparison of this result with the steam consumption previous to the installation of the turbine, when the load was carried by two independent compound-condensing engines, it was necessary to add 10 to 15 per cent. to the foregoing figures to compensate for the variable load, as the comparative data were obtained from a 24-hr. test of the complete plant on Sept. 13 to 14, 1911. This test showed a steam consumption of 26.55 lb. per kilowatt-hour for both engines taken together carrying an average of 992.7 equivalent kilowatts for the 24-hour test period.

Adding 15 per cent. for the variable load on a 24-hour test, to 16.95 lb., the result obtained on the steady load combined engine and turbine test gives 19.49 lb. per kilowatt-hour for the comparative steam consumption as against 26.55 lb. for the two engines. This shows a gain of 7.06 lb. per kilowatt-hour, or 26.6 per cent. in favor of the turbine and engine.

The cost of the 1500-kw. low-pressure turbine installation was $38,300, including the condenser, piping, separators, expansion joints, special valves, foundations, etc., but not including the switchboard or the building. This is equivalent to about $25 per kilowatt. The cost of the 2500-hp. noncondensing engine installation was $32,000, including the piping, separator, valves, foundations, etc., but not including the building, making about $13 per horsepower.

This makes the installation cost of the combined engine and low-pressure turbine unit $70,300. The fixed charges on this amount, taking interest at 5 per cent., depreciation at 5 per cent., taxes at 1½ per cent. and insurance at 1 per cent., or a total of 12½ per cent., would be $8787 per year. The cost of operating the plant with two condensing engines during the year 1911 was $63,457. Assuming that there would have been an increase of one-third in this operating cost, during the present year, due to increased demand for power, under the old conditions, this would amount to $84,609 for the probable operating cost for 1912. Taking an average yearly saving of 20 per cent. due to the new installation of an engine and low-pressure turbine, based on the 26.6 per cent. saving as shown by the tests referred to, there would be $16,920 as the probable yearly saving due to the new machinery or nearly twice the fixed charges of $8787 on the entire installation cost.

Comparing the cost of the engine and low-pressure turbine unit as installed with the cost of a high-pressure tur-

bine installation of similar capacity shows the following: A high-pressure turbine rated at 2000 kw. with 25 per cent. continuous overload capacity and 50 per cent. overload for two hours, with 175 lb. steam pressure, no superheat, a 28-in. vacuum and running at 1800 r.p.m., would have very nearly the same equivalent capacity as the engine and low-pressure turbine unit under discussion; if anything, the capacity would be less than the combined unit. The installed cost of such a high-pressure unit would be about $45,500, including the condenser, piping, separator, expansion joints, foundations, etc., but not including the switchboard or building so as to correspond with the costs of the combined unit. This cost is based on actual proposals for turbine, condenser, etc., and should be very close to the actual cost under the conditions at this particular plant.

Comparing this figure with the actual cost of the engine and low-pressure turbine, $70,300, shows an increased cost of $24,800 for the combination, and taking 12½ per cent. of this gives $3100 per year for fixed charges on the additional investment required.

A comparison in steam consumption between the engine and low-pressure turbine combination and the high-pressure turbine alone as heretofore given, shows a lower consumption at full load for the combined unit by 8.4 per cent. If it is assumed that this difference would exist at the average load carried during the year, which would probably be somewhere between three-quarters and full load, and taking the yearly cost of operation at $84,609 as before, 8.4 per cent. of this cost, or $7107, would represent the yearly saving in favor of the engine and low-pressure turbine unit. This is more than twice the yearly fixed charges of $3100 on the additional investment required for the latter.

In conclusion, it may be said, that from the results of the tests and comparisons given above, the conclusion would be warranted that there is sufficient margin between the steam consumption of a unit composed of a noncondensing engine and low-pressure turbine as compared with a high-pressure turbine, under the conditions described, to unquestionably throw the decision in favor of the engine and low-pressure turbine and warrant the additional expenditure required for them.

Steam Leakage Past Piston Valves

The Feb. 9 issue of *The Engineer* contains a very interesting account of some investigations recently conducted by H. D. Lobley at the University of Birmingham to determine the loss of steam due to leakage past a piston valve.

A striking feature of these tests is that the results are quite contrary to those obtained by Callendar and Nicolson in a similar series of tests carried out in 1897. Mr. Lobley's results indicate that the loss due to leakage past a piston valve is comparatively small, whereas the conclusion from Callendar and Nicolson's experiments was that this loss formed a large part of the "missing quantity," a term used to designate the thermal losses within the steam-engine cylinder.

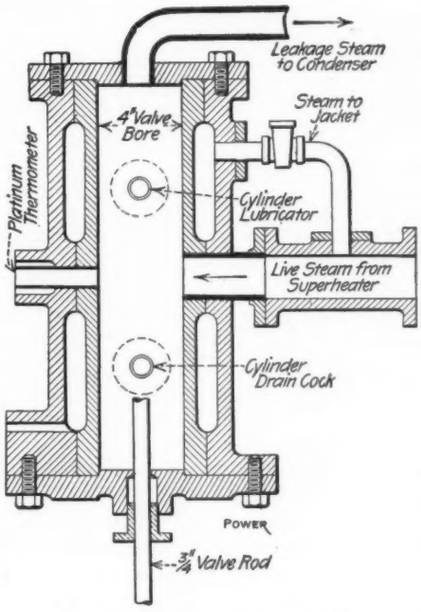
The experiments as conducted by Mr. Lobley were made with an ordinary 4-in. piston valve fitted with four ramsbottom spring rings and taken from stock. The valve was fitted into a jacketed cylinder and was driven at various speeds by a motor. Steam was supplied from the boiler or superheater to the center of the valve and the leakage was taken from the end. This steam was led to a condenser and after being condensed was carefully weighed. The illustration shows a section through the cylinder. A steady flow of steam was allowed to blow through the jacket so as to prevent stagnation and a corresponding reduction in temperature which probably would have occurred if the amount of steam used had been only that due to leakage.

During each of the tests, readings were taken every 15 min. The first run was made with superheated steam, a boiler pressure of 200 lb. per square inch and variable speeds; the second was at the same steam pressure with a constant speed and at different degrees of superheat; the third was with saturated steam at different pressures and a constant speed. The results are shown in the accompanying tables.

The loss due to leakage alone depends upon the pressure difference and the speed; also, it is much less where superheated steam is used.

The total leakage, however, was found to be comparatively small.

As a means of comparison with Callendar and Nicolson's results, Mr. Lobley applies his own figures to the formula deduced by them and then solves for the leakage coefficient. This expression is as follows:



SECTION THROUGH VALVE CHEST

$$K = \frac{CP}{L}$$

where,

K = Rate of leakage per hour per pound difference of pressure;

P = Perimeter of port;

L = Mean overlap of valve;

C = Leakage coefficient.

Although these terms were originally applied to a slide valve, in the case of a piston valve, P would represent the circumference of a ring and L would be the sum of the widths of the rings, representing the length of the contact surface. The assumption was made that the leakage past the two portions of the valve was equal.

Table 4 shows the values of C calculated from the saturated-steam tests with variable pressure differences.

TABLE 1. SUPERHEATED STEAM

Trial	Speed in R.p.m.	Temperature of Steam, Deg. F.	Boiler Gage Pressure, Lb. per Sq.In.	Leakage in Lb. per Hour
A 1..	200	407	200	4 825
B 2..	150	396	200	4 82
B 3..	100	392	200	4 9
B 4..	50	397	200	4 8
B 5..	(standing)	409	200	2.4

TABLE 2. SUPERHEATED STEAM

Trial	Speed in R.p.m.	Temperature of Steam, Deg. F.	Boiler Gage Pressure, Lb. per Sq.In.	Leakage in Lb. per Hour
A 1..	200	407	200	4 825
A 2..	200	452	200	3 66
A 3..	200	498	200	2.70
A 4..	200	557	200	3.075
A 5..	200	588	200	3 275

TABLE 3. SATURATED STEAM

Trial	Pressure in Lb. per Sq.In.	Speed in R.p.m.	Leakage in Lb. per Hour
C 1...	156.1	200	7.8
C 2...	128	200	7.5
C 3...	80	200	6.6
C 4...	37.4	200	4.9
C 5...	152	(standing)	3.86

From this an average value of 0.0022 is obtained for C, or about one-tenth of that found by Callendar and Nicolson. The value of C, however, is not constant,

TABLE 4. LEAKAGE COEFFICIENT

Pressure Difference	Leakage in Lb. per Hour	K	C
141	7 8	0.0276	0.0011
113	7 5	0.03325	0.00132
65	6 6	0.05075	0.00202
22 4	4 9	0.1095	0.00436

although it does not vary directly with the pressure difference. The relative amount of this leakage, according to Mr. Lobley's results, may be arrived at closely by considering that a 4-in. piston valve would handle about 1500 lb. of steam per hour. The valve when running at a temperature of 557 deg. F. leaks only 3.075 lb. of steam per hour. This represents only about 0.2 per cent. of the steam handled.

The effect of temperature upon the leakage is clearly shown by the tests. By slightly superheating the steam, the leakage was reduced from 7.8 to 4.8 lb. per hour.

An explanation of this is given, based upon the condensation and reëvaporation theory. This is to the effect that the space between the two portions of the valve is constantly in contact with steam at a high temperature while the exhaust side is exposed to the comparatively cold steam at atmospheric pressure. When the valve begins to move it exposes a cold portion of the liner and some hot steam is immediately condensed. The film of water thus formed allows the rings to pass over it, and upon emerging on the exhaust side of the valve it evaporates again. With superheated steam, the cold walls are heated to a large extent at the expense of the superheat and much less condensation results.

Chimneys for Dissipating Smoke

By Dr. Alfred Gradenwitz

A new arrangement effecting a thorough dilution of smoke and air has been invented by Dr. H. Wislicenus and is being promoted by a German metallurgical firm (Metallbank und Metallurgische Geselleschaft, at Frankfort-on Main). This "dissipator," as it is called, is based on the following principle:

Instead of escaping from the top of the chimney in the usual manner, the exhaust gases are caused to leave through a number of approximately horizontal air channels, proportioned to their mass and the rate of flow. The gases are thus whirled through with the air current within the dissipator chimney and in its immediate neighborhood, without affecting the draft.

The upper part of the chimney, one-fourth to one-third of its height, is latticed (see Fig. 1). In this section the air currents are mixed on one side in the funnel-shaped or slot-like channels with the rising current of exhaust gases, the air currents deviating and changing their speed so as to produce a whirling action. The mixture thus formed escapes through the air channels on the opposite side. Only a small part of the gases escapes through each series of channels, the remainder continuing to rise, and becoming more and more diluted before escaping through the slots. The air channels are widened out conically, which design is especially favorable for the formation of whirls and for insuring a thorough mixture. All heavy smoke, with a strong wind, is thus dissolved into a foggy mist, while with a light wind it is at least thoroughly distributed and diluted.

The shape and dimensions of the air channels can be varied in many ways so as to further assist the dissipator action; by a tangential arrangement (Fig. 3), by a double funnel-shaped air channel (Fig. 4) or by an extension of part of the channels into diverging passages (Fig. 5). All these modifications have been protected by letters patent.

Whenever in the place of a draft chimney, a channel mouth terminating on the level or on some roof or hill has to be dealt with, the discharge shaft can be designed with multiple gratings, insuring a more thorough mixture (see Fig. 6). The first instance of such a "multi-dissipator" chimney is a stack erected in connection with a Winkler washing chamber (for removing acids), while an improved plant consisting of radial bricks —with a moist ventilator and limestone plant for the removal of acids—was installed for a large German dyeing and bleaching works. These chimneys, on account of the convenient distribution of their slots, are of a pleasing appearance, and their stability has been found

Fig. 1. Dissipator Stack Compared with Ordinary Chimney When Handling Moderate Amount of Smoke

Fig. 2. Stack Handling Heavy Tar Smoke, Notice How Smoke Is Diluted

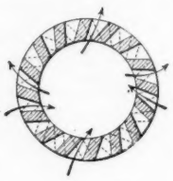


Fig. 3

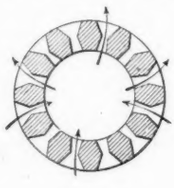


Fig. 4

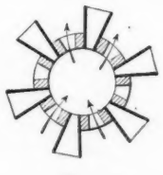


Fig. 5

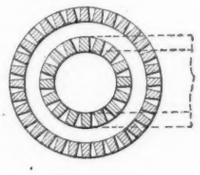


Fig. 6

Different Forms of Dissipator Chimneys

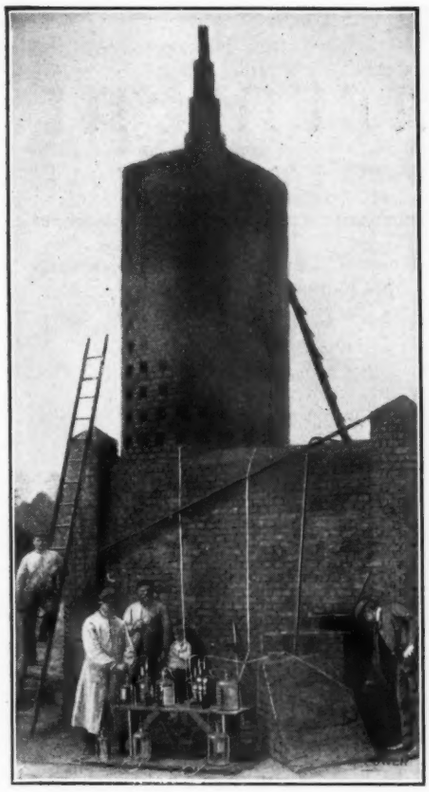


FIG. 7. FIRST EXPERIMENTAL MULTI-DISSIPATOR STACK, SHOWING SMOKE DILUTED INTO A HAZE

to be quite as satisfactory as that of ordinary stacks.

Fig. 7 shows the first multi-dissipator constructed by Dr. Wislicenus and illustrates how the smoke is dissolved into a foggy mist.

"Albany" Shaking Grate

George H. Thacher & Co., Albany, N. Y., manufacture the "Albany" shaking grate herewith illustrated. For the larger sizes of furnace the grate area is divided into four quarters, each of which shakes as a unit as indicated in Fig. 1. All parts are standardized and completely interchangeable, there being no right- or left-hand grates. The grate bars are triangular in shape which, it is claimed, gives the best distribution of metal and the greatest strength and durability.

The individual parts of the grate are shown in Fig. 3, while Fig. 2 shows them assembled. The bearing bar *A* is set in the brickwork and carries the side frame *B* as shown. The end pieces *C* fit into the wedge-shape slots in the side frames. The grate-bar frames *D* are carried in bearings in the side frames and are joined together underneath by the connecting bar *E*, to which the connecting-rod *K* is attached. The grate surface proper is composed of a number of grate sections *O* which fit into the grate-bar frame and are fastened in place by a split pin which passes through the lug on the section and the bottom member of the frame, as shown in the lower part of Fig. 1. The advantage of this construction is that the amount of air space of any installation may be changed simply by changing the grate sections *O* without making any change in the rest of the installation and when a section burns out only that section has to be replaced and not a whole bar. These grate sections are made in several sizes to give any percentage of air space from a 1/8-in. opening for fine fuels up to a 3/4-in. opening, for bituminous lump coal. In addition, sections with pinhole openings may be obtained for especially fine fuels.

The journals at the ends of the grate-bar frames are flattened on the top and a section somewhat smaller in width than the diameter of the journal is cut out of the top part of the bearings. Thus, any frame may be lifted out of its bearings and removed entirely by simply disconnecting it from the connecting bar and turning the frame at an angle of 90 deg.

During the shaking operation of the grate, when the sections are tilted, a wedge-shape pocket is formed by the top of one section and the side of the next into which the under stratum of ash is pushed by the first motion of the grate. When the sections are again brought back to the normal position, this ash is cut off from the live fuel and dropped into the ashpit. This pocket-forming arrangement tends to prevent the loss of unburnt fuel to the ashpit.

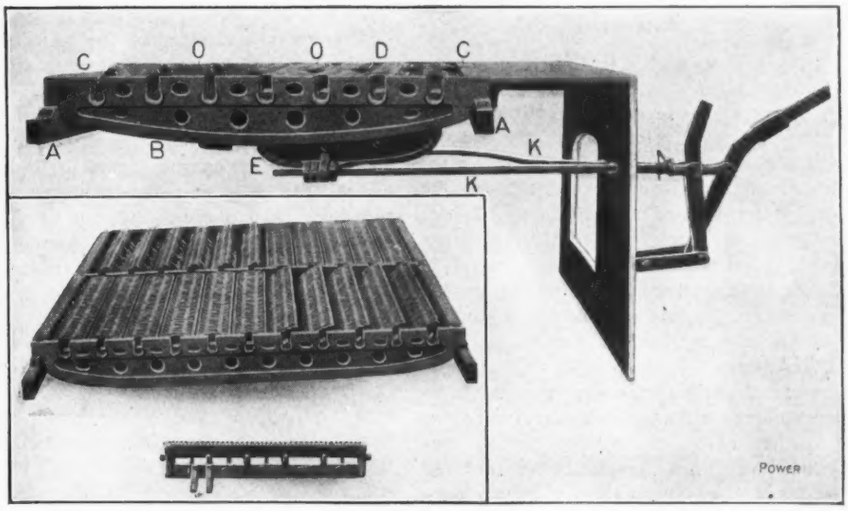


FIG. 1. THE "ALBANY" SHAKING GRATE FIG. 2. SHOWING ASSEMBLY OF PARTS

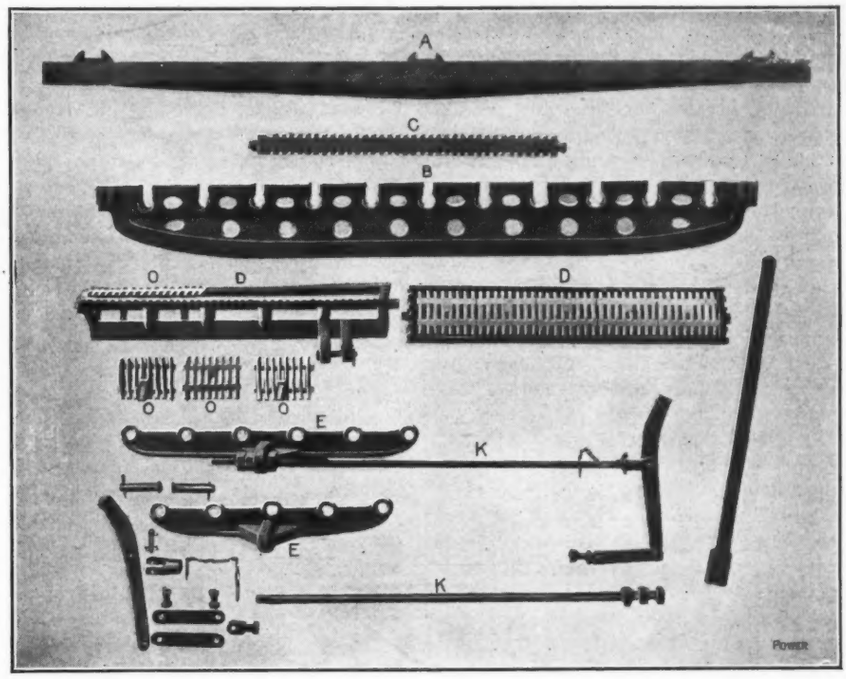


FIG. 3. SHOWING INDIVIDUAL PARTS OF THE "ALBANY" SHAKING GRATE

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Wires for Direct-current Circuits

By Cecil P. Poole

There are two ways of getting at the proper size of wire to use for electric circuits in order to keep the line loss within desired limits; one by considering the power transmitted and the other by considering the current strength, regardless of the power. The former method is more easily applied in alternating-current work, but the latter is simpler of application in direct-current work.

No matter which method may be used, however, the current per wire must be considered, either directly or indirectly, in the case of indoor wiring because the fire insurance authorities will not allow more than a certain current to be carried by each standard size of wire, in order to guard against overheating. Table 1 specifies the maximum allowable current for every standard size of copper wire in regular use for power and lighting circuits in the United States. It also shows what the drop in voltage is per foot of each size of wire, for each ampere of current carried; the drop per foot of distance (for a two-wire circuit) per ampere of current; the drop in volts and the loss in watts per foot of distance when carrying the greatest allowable current, and the drop and lost watts per foot of distance when carrying four-fifths of the greatest allowable current. This last group of figures applies essentially to motor circuits and shows the greatest drop and the greatest loss per foot of distance that are allowable under the insurance rules. The rules require that wires used for connecting motors to supply circuits shall be large enough to carry 25 per cent. more than the full-load current of the motors.

For example, No. 4 wire with rubber insulation is allowed to carry only 65 amperes, maximum. Therefore, a motor connected up with this size of wire must not take more than 80 per cent. of 65 amperes ($0.8 \times 65 = 52$ amperes) at full load, because the 25 per cent. margin required will make the maximum current $52 \times 1.25 = 65$ amperes. Neither the drop nor the loss at 65 amperes, in such a case, is important, because that current would exist only for a few moments while starting the motor and perhaps occasionally when momentary overloads occur. It is the full-load drop and loss that are of importance; that is why columns 10, 11, 14 and 15 were put in Table 1.

The usefulness of these columns lies in showing by a single simple multiplication whether a given size of wire can be used for a given drop. For example, suppose a 220-volt motor is to be put in at a distance from the mains of 75 ft., and a drop of 1 per cent. is desired at full load. Also suppose that you have on hand a lot of No. 4 and No. 6 rubber-covered wire but no other near sizes. At 1 per cent. drop over a distance of 75 ft. the drop per foot would be

$$\frac{0.01 \times 220}{75} = 0.0293$$

of a volt. Reference to the tenth column of the table shows that the greatest allowable drop for No. 4 is 0.0263 of a volt and that for No. 6 is 0.0296; therefore, the No. 6 wire can be used.

The drop columns are useful also in determining the limits of drop in another way. For example, suppose a 250-volt motor requiring 60 amperes at full load is to be installed at a distance of 150 ft. from the mains and the drop must not exceed 2 per cent., which would be 5 volts. What size of rubber-insulated wire should be used? According to the fourth column of the table, No. 3 wire could be used without exceeding the insurance limits, because 60 amperes plus 25 per cent. would make the maximum current 75 amperes and the insurance rules allow 76. But would the drop come within the limit of 2 per cent.? According to column 10, the allowable full-load drop per foot of distance is 0.0244 of a volt, so that for 150 ft. the full-load drop would be $150 \times 0.0244 = 3.66$ volts which comes well within the specified limit of 5 volts.

Columns 12 to 15 correspond to columns 8 to 11; they give the loss in watts under the same conditions for which columns 8 to 11 give the drop in volts. This is useful in cases where the current is not stated but the power is. For example, if it were known that 39 kw. had to be transmitted 260 ft. from a 220-volt line with a loss of not more than 2 per cent. it would not be necessary to divide 39,000 by 220 to find the line current; then see how small a size of wire would satisfy the fire-insurance rules, and finally figure the drop to see if it came within the 2 per cent. limit. All that is required is to divide 2 per cent. of 39,000 by 260 to get the loss in watts per foot of distance and find the nearest larger number in column 12; this number is 3.22 and the size of wire at the left-hand end of the same line of type is No. 0. Therefore, No. 0 can be used to satisfy both the insurance and the loss requirements.

Column 8 would not apply to this case without checking up the insurance requirements separately. Thus, the loss being 2 per cent. or less, the drop will be 2 per cent. or less. Now 2 per cent. of 220 volts is 4.4 volts, and this would be a drop of $4.4 \div 260 = 0.0169$ volt per foot. Column 8 gives no intimation as to whether one should use wire of 400,000 circ.mils cross-section, with a maximum drop of 0.0174 volt per foot of distance, or some of the smaller sizes, which are all capable of a higher drop per foot of distance. The *current* must be known in order to use columns 7 to 11. These columns are very useful, however, when the current is known, and it usually is.

Columns 8 to 15, although highly convenient in cases where the current is near the maximum allowed by the insurance rules for one of the standard sizes of wire, are not of much service in dealing with currents which fall between the insurance limits for neighboring wire sizes. For example, with a current of 64 or 65 amperes, this table shows at once whether No. 4 rubber-insulated (or No. 6 weatherproof) wire will give the desired drop or loss. But if the full-load current of a motor happened to be, say, 48 amperes, Table 1 would not show what the drop would be with any size of wire allowed by the insurance rules, except by the use of the figures in column 7, which involves at least two probably tedious calculations. Adding the 25 per cent. margin required would carry the current up to 60 amperes maximum allowance. With rubber-covered wire, any size larger than No. 5 could be used, and the drop would be less than 0.0263 volt per foot of distance; with No. 4 it would be $\frac{60}{65}$ of 0.0263, with No. 3 it would be $\frac{60}{75}$ of 0.0263, with No. 2 $\frac{60}{80}$ or $\frac{3}{4}$ of 0.0263, and so on. The determination of the drop, however, requires a good deal of tiresome arithmetic, applied on the cut-and-try plan.

In order to avoid this, with the attendant possibilities of arithmetical error due to haste or the misplacement of a decimal point, Table 2 is provided. This shows for each size of wire the product of amperes $\times$ feet corresponding to different drops in voltage from $\frac{1}{4}$ to 60 volts. For example, if the cur-

rent multiplied by the distance were equal to 50,000, the drop with No. 0000 wire would be 5 volts; with No. 0 wire 10 volts; with No. 2 wire about 16 volts; with No. 5 wire 32 volts, etc. The table was computed on a calculating machine and the number of figures in each of the large numbers is therefore greater than is really necessary.

Finding the Size of Wire

For finding the proper size of wire to use for a given load, distance and drop, multiply the current by the distance and in the line with the desired drop, find the nearest number to the product of current × feet; at the head of the column in which that number is located will be found the size of wire.

Example I. A pair of wires 180 ft. long must carry 100 amperes with a drop of about 3 volts; what size of wire should be used?

The distance being 180 ft. and the current 100 amperes, the product (amperes × feet) is 18,000. The drop is to be 3 volts; starting at 3 in the drop column and following along that line, the nearest number to 18,000 ampere-feet is 18,915, which is located in the fourth column of the table, and the size of wire at the top of that column is No. 00. Now, this size must be checked for insurance limitation. Table 1 shows that No. 00 is allowed to carry 150 amperes with rubber insulation or 220 with other kinds, so that it is within the limit, even for motor service.

Example II. A 250-volt motor requiring 75 amperes at full load must be connected to a supply circuit 700 ft. away, and as the connecting wires are out of doors, weatherproof wire can be used. It is desired that the drop shall not be more than 4 per cent., which, of course, is 10 volts. What size of wire should be used?

The current × distance will be 52,500 and starting at 10 volts in the drop column it will be found that 50,000 is the nearest number to 52,500; it is located in the No. 0 column and immediately below it is the number 55,000, which is in the line with 11 volts drop. Therefore, if No. 0 wire be used, the drop will be half-way between 11 volts and 10 volts, or 10½. If it is important to keep the drop within 10 volts, No. 00 wire must be used. The nearest number to 52,500 in this column is 50,440, in the line with 8 volts drop; the actual drop with this size of wire, therefore, would be between 8 and 9 volts, because the amperes × feet are 52,500, whereas with No. 00 wire they would have to be 50,440 to give 8 volts drop or 56,745 to give 9 volts drop.

Finding the Drop

From the last example it should be clear that when the current, distance and wire size are known the drop can be found, at least approximately, from Table 2 by multiplying the current and distance together, finding the nearest number to the product in the column under the wire size and following along that line to the drop column.

Example III. A pair of No. 8 wires 25 ft. long carry 28 amperes; what is the drop?

The current × distance = 700; the nearest number to this in the No. 8 column of Table 2 is 782 and this is in

TABLE 1. PROPERTIES OF COPPER WIRES

Brown & Sharpe Gage

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
			Greatest Allowable Current*				Volts drop per foot of distance,† carrying maximum current.		Volts drop per foot of distance† with 80% of maximum current.		Watts lost per foot of distance,† with maximum current.		Watts lost per foot of distance;† 80% of max. current.	
Wire Size; No.	Diameter in Mils	Area in Circular Mils	Rubber Insul.	Other Insul.	Volts Drop per Ampere per Foot of Wire	Volts Drop per Ampere per Foot of Distance†	Rubber Insulation	Other Insulation	Rubber Insulation	Other Insulation	Rubber Insulation	Other Insulation	Rubber Insulation	Other Insulation
		2,000,000	1050	1670	0.00000528	0.00001056	0.01109	0.01763	0.00887	0.01411	11.64	29.44	7.45	18.84
		1,900,000	1010	1610	0.00000556	0.00001112	0.01123	0.01790	0.00900	0.01432	11.34	28.81	7.26	18.44
		1,800,000	970	1550	0.00000586	0.00001173	0.01138	0.01818	0.00910	0.01455	11.04	28.17	7.06	18.03
		1,700,000	930	1490	0.00000621	0.00001242	0.01155	0.01851	0.00924	0.01480	10.74	27.58	6.87	17.65
		1,600,000	890	1430	0.00000660	0.00001320	0.01175	0.01888	0.00940	0.01510	10.45	27.00	6.69	17.27
		1,500,000	850	1360	0.00000704	0.00001408	0.01197	0.01915	0.00957	0.01532	10.17	26.04	6.51	16.67
		1,400,000	810	1290	0.00000754	0.00001508	0.01221	0.01945	0.00977	0.01556	9.89	25.09	6.33	16.05
		1,300,000	770	1220	0.00000812	0.00001624	0.01250	0.01981	0.01000	0.01585	9.62	24.16	6.15	15.46
		1,200,000	730	1150	0.00000880	0.00001760	0.01285	0.02024	0.01028	0.01619	9.38	23.27	6.00	14.89
		1,100,000	690	1080	0.00000960	0.00001920	0.01325	0.02074	0.01060	0.01659	9.14	22.40	5.85	14.33
		1,000,000	650	1000	0.00001056	0.00002112	0.01373	0.02112	0.01098	0.01689	8.92	21.12	5.71	13.51
		900,000	600	920	0.00001173	0.00002346	0.01408	0.02158	0.01126	0.01726	8.44	19.85	5.40	12.70
		800,000	550	840	0.00001320	0.00002640	0.01452	0.02218	0.01162	0.01774	7.97	18.63	5.10	11.92
		700,000	500	760	0.00001510	0.00003020	0.01510	0.02295	0.01208	0.01836	7.55	17.44	4.83	11.16
		600,000	450	680	0.00001760	0.00003520	0.01584	0.02394	0.01267	0.01915	7.12	16.28	4.55	10.42
		500,000	390	590	0.00002112	0.00004225	0.01648	0.02493	0.01316	0.01994	6.42	14.70	4.11	9.41
		400,000	330	500	0.00002640	0.00005280	0.01742	0.02640	0.01394	0.02112	5.74	13.20	3.67	8.44
		300,000	270	400	0.00003520	0.0000704	0.01900	0.02816	0.01521	0.02253	5.13	11.26	3.28	7.21
		200,000	200	300	0.00005278	0.0001056	0.02112	0.0317	0.01690	0.02534	4.22	9.51	2.70	6.08
0000	460.0	211,600	210	312	0.0000499	0.000100	0.02100	0.0312	0.01680	0.02496	4.41	9.73	2.82	6.23
000	409.6	167,770	177	262	0.0000630	0.000126	0.02230	0.0330	0.01784	0.02641	3.94	8.64	2.52	5.53
00	364.8	133,000	150	220	0.0000793	0.000159	0.02385	0.0350	0.01908	0.02798	3.57	7.70	2.29	4.92
0	324.9	105,560	127	185	0.000100	0.000200	0.02540	0.0390	0.02032	0.03120	3.22	7.21	2.06	4.61
1	289.3	83,690	107	156	0.000126	0.000252	0.0270	0.0393	0.02157	0.03145	2.88	6.13	1.84	3.92
2	257.6	66,360	90	131	0.100159	0.000318	0.0286	0.0417	0.0229	0.03332	2.57	5.46	1.64	3.49
3	229.4	52,625	76	110	0.000201	0.000402	0.0306	0.0442	0.0244	0.03537	2.32	4.86	1.48	3.11
4	204.3	41,740	65	92	0.000253	0.000506	0.0329	0.0465	0.0263	0.0372	2.13	4.27	1.36	2.73
5	181.9	33,090	54	77	0.000319	0.000638	0.0345	0.0491	0.0276	0.0393	1.86	3.78	1.19	2.42
6	162.0	26,245	46	65	0.000402	0.000804	0.0370	0.0523	0.0296	0.0418	1.70	3.40	1.09	2.17
8	128.5	16,512	33	46	0.000639	0.001278	0.0422	0.0588	0.0337	0.0470	1.39	2.70	0.889	1.73
10	101.9	10,384	24	32	0.001017	0.002034	0.0488	0.0651	0.0390	0.0521	1.17	2.08	0.749	1.33
12	80.8	6,530	17	23	0.001617	0.003234	0.0550	0.0744	0.0440	0.0595	0.935	1.71	0.598	1.09
14	64.1	4,107	12	16	0.00257	0.00514	0.0617	0.0822	0.0493	0.0658	0.740	1.31	0.473	0.842
16	50.8	2,583	6	8	0.00409	0.00818	0.049	0.0654	0.0393	0.0523	0.294	0.523	0.188	0.334
18**	40.3	1,624	3	5	0.0065	0.013	0.039							

*The limit (in amperes) allowed by the fire insurance rules for indoor wiring. For outdoor overhead wires, twice these figures may be allowed without danger of overheating.

†For a two-wire circuit.

**This size is allowed indoors only in flexible lamp cords.

the same line with 1 volt; the drop, therefore, will be between 1 volt and ¾ volt —nearer 1 volt.

Example IV. An outdoor line of No. 2 weatherproof wire carries 200 amperes 140 ft. from a 230-volt circuit to a factory building containing lamps and motors; what is the voltage at the entrance to the building?

The current × distance = 28,000. The nearest number to this in the No. 2 wire column of Table 2 is 28,300, which is in the line with 9 volts. Therefore, 28,300 amperes × feet would give 9 volts drop and 28,000 will give $\frac{28{,}000}{28{,}300}$ of 9 volts, or 8.9 volts drop. The voltage at the factory building, therefore, will be 230 — 8.9 = 221.1 volts.

Finding the Practical Current

It sometimes happens that an old existing feeder or distribution main can be put into use to carry part of an extension load or for some other purpose, and it is desired to know about how much load it will be practical to put on it. The first step in such a case would be to find out whether the line will carry the full insurance allowance without too much drop. This is more quickly done with Table 1, using the drop columns at the right. If the drop is too great with the current allowed, then Table 2 will be more convenient for finding how much current can be carried without too great a drop.

Example V. It is desired to load a pair of No. 4 rubber-covered wires 100 ft. long as heavily as possible without making the drop exceed 2½ volts; how much current will the line carry?

The allowable drop is 2½ volts and the distance 100 ft.; the drop per foot, therefore, must not exceed

$$\frac{2\frac{1}{2}}{100} = 0.025$$

of a volt. The eighth column of Table 1 shows that the drop per foot of distance in a pair of No. 4 rubber-insulated wires carrying maximum current is 0.0329 of a volt, which is much beyond the desired limit of 0.025 of a volt per foot.

The required current can be found quickly from Table 2. In the line with 2½ volts drop and the column under No. 4 wire is the number 4940. This is amperes × feet, so that dividing it by the

TABLE 2. RELATION BETWEEN DROP AND CURRENT × DISTANCE

The numbers in the body of the table are amperes × feet for a two-wire circuit

Volts Drop	Size of Wire, B & S Gage														
	0000	000	00	0	1	2	3	4	5	6	8	10	12	14	16
½	2,505	1,984	1,576	1,250	990	786	622	494	392	311	195	123	77	48	30
¾	5,010	3,968	3,152	2,500	1,980	1,572	1,243	988	783	622	390	246	154	97	61
¾	7,515	5,952	4,728	3,750	2,975	2,358	1,865	1,482	1,175	933	586	369	231	145	91
1	10,020	7,936	6,305	5,000	3,968	3,145	2,487	1,976	1,567	1,244	782	492	309	194	122
1¼	12,525	9,920	7,880	6,250	4,960	3,930	3,109	2,470	1,959	1,555	978	615	386	243	152
1½	15,030	11,900	9,457	7,500	5,950	4,717	3,731	2,964	2,351	1,866	1,174	737	463	292	183
1¾	17,535	13,890	11,034	8,750	6,945	5,503	4,353	3,458	2,742	2,177	1,369	860	541	340	213
2	20,040	15,870	12,610	10,000	7,936	6,289	4,975	3,952	3,134	2,487	1,565	983	618	389	244
2¼	22,545	17,855	14,185	11,250	8,927	7,075	5,595	4,445	3,525	2,799	1,761	1,106	695	437	274
2½	25,050	19,840	15,760	12,500	9,920	7,861	6,219	4,940	3,918	3,109	1,956	1,229	773	486	305
2¾	27,555	21,825	17,340	13,750	10,910	8,647	6,840	5,430	4,310	3,420	2,152	1,352	850	535	335
3	30,060	23,810	18,915	15,000	11,904	9,434	7,462	5,928	4,700	3,731	2,347	1,475	927	583	366
3½	35,070	27,775	22,065	17,500	13,885	11,005	8,706	6,915	5,485	4,353	2,739	1,721	1,082	680	427
4	40,080	31,745	25,220	20,000	15,870	12,575	9,950	7,905	6,269	4,975	3,130	1,967	1,236	778	488
4½	45,090	35,710	28,370	22,250	17,855	14,150	11,190	8,890	7,050	5,597	3,521	2,212	1,391	875	549
5	50,100	39,680	31,525	25,000	19,840	15,720	12,437	9,880	7,837	6,219	3,912	2,458	1,546	972	610
5½	55,110	43,650	34,675	27,500	21,825	17,295	13,680	10,870	8,620	6,841	4,304	2,704	1,700	1070	671
6	60,120	47,620	37,830	30,000	23,810	18,868	14,925	11,855	9,400	7,463	4,695	2,950	1,855	1167	732
6½	65,130	51,585	40,980	32,500	25,790	20,440	16,170	12,845	10,185	8,085	5,086	3,196	2,009	1264	793
7	70,140	55,555	44,135	35,000	27,775	22,010	17,410	13,830	10,970	8,706	5,477	3,441	2,164	1361	854
7½	75,150	59,520	47,285	37,500	29,760	23,585	18,655	14,820	11,755	9,328	5,868	3,687	2,319	1459	915
8	80,160	63,490	50,440	40,000	31,745	25,155	19,900	15,810	12,540	9,950	6,260	3,933	2,473	1556	976
9	90,180	71,425	56,745	45,000	35,710	28,300	22,385	17,785	14,100	11,194	7,042	4,425	2,782	1750	1098
10	100,200	79,365	63,050	50,000	39,680	31,445	24,875	19,760	15,670	12,438	7,825	4,916	3,092	1945	1221
11	110,220	87,300	69,355	55,000	43,650	34,590	27,360	21,740	17,240	13,682	8,607	5,408	3,401	2140	1343
12	120,240	95,235	75,660	60,000	47,620	37,735	29,850	23,715	18,800	14,925	9,390	5,900	3,710	2334	1465
13	130,260	103,170	81,965	65,000	51,585	40,880	32,335	25,690	20,375	16,169	10,172	6,391	4,019	2529	1587
14	140,280	111,110	88,270	70,000	55,555	44,025	34,025	27,665	21,940	17,413	10,955	6,883	4,329	2723	1709
15	150,300	119,045	94,575	75,000	59,520	47,170	37,310	29,640	23,510	18,657	11,737	7,375	4,638	2918	1831
16	160,320	126,980	100,880	80,000	63,490	50,310	39,800	31,620	25,075	19,900	12,520	7,866	4,947	3112	1953
17	170,340	134,920	107,185	85,000	67,460	53,460	42,285	33,595	26,645	21,144	13,302	8,358	5,256	3307	2075
18	180,360	142,855	113,490	90,000	71,425	56,600	44,775	35,570	28,210	22,388	14,085	8,850	5,565	3501	2197
19	190,380	150,790	119,800	95,000	75,395	59,745	47,260	37,550	29,780	23,632	14,867	9,341	5,875	3696	2319
20	200,400	158,730	126,100	100,000	79,365	62,890	49,750	39,525	31,345	24,876	15,649	9,833	6,184	3891	2442
21	210,420	166,665	132,405	105,000	83,330	66,035	52,235	41,500	32,915	26,119	16,432	10,324	6,493	4085	2564
22	220,440	174,600	138,710	110,000	87,300	69,180	54,725	43,475	34,480	27,363	17,214	10,816	6,802	4280	2686
23	230,460	182,540	145,015	115,000	91,270	72,325	57,210	45,450	36,050	28,607	17,997	11,308	7,111	4474	2808
24	240,480	190,475	151,320	120,000	95,235	75,470	59,700	47,430	37,615	29,851	18,779	11,799	7,421	4669	2930
25	250,500	198,410	157,630	125,000	99,205	78,615	62,190	49,400	39,180	31,095	19,562	12,291	7,730	4863	3052
26	260,520	206,350	163,930	130,000	103,175	81,760	64,675	51,380	40,750	32,338	20,344	12,783	8,039	5058	3174
27	270,540	214,285	170,240	135,000	107,140	84,905	67,160	53,360	42,320	33,582	21,127	13,274	8,348	5252	3296
28	280,560	222,220	176,540	140,000	111,110	88,050	69,650	55,335	43,885	34,826	21,909	13,766	8,658	5447	3418
29	290,580	230,155	182,850	145,000	115,075	91,195	72,135	57,310	45,450	36,070	22,692	14,258	8,967	5642	3540
30	300,600	238,095	189,155	150,000	119,045	94,340	74,625	59,285	47,020	37,313	23,474	14,749	9,276	5836	3663
32	320,640	253,965	201,765	160,000	126,980	100,625	79,600	63,240	50,155	39,800	25,039	15,733	9,894	6225	3907
34	340,680	269,840	214,375	170,000	134,920	106,915	84,575	67,190	53,290	42,289	26,604	16,716	10,513	6614	4151
36	360,720	285,710	226,985	180,000	142,855	113,205	89,550	71,145	56,425	44,776	28,169	17,699	11,131	7003	4395
38	380,760	301,585	239,595	190,000	150,790	119,495	94,525	75,100	59,560	47,264	29,734	18,682	11,750	7392	4639
40	400,800	317,460	252,205	200,000	158,730	125,785	99,500	79,050	62 695	49,751	31,299	19,666	12,368	7782	4884
42	420,840	333,330	264,815	210,000	166,665	132,075	104,475	83,000	65,830	52,239	32,864	20,649	12,987	8171	5128
44	440,880	349,205	277,425	220,000	174,600	138,360	109,450	86,950	68,965	54,726	34,429	21,632	13,605	8560	5372
46	460,920	365,080	290,035	230,000	182,540	144,650	114,425	90,900	72,100	57,214	35,994	22,616	14,223	8949	5616
48	480,960	380,950	302,645	240,000	190,475	150,940	119,400	94,860	75,235	59,700	37,559	23,599	14,842	9338	5860
50	501,000	396,825	315,255	250,000	198,410	157,230	124,375	98,810	78,370	62,189	39,124	24,582	15,460	9727	6105
52	521,040	412,700	327,865	260,000	206,350	163,520	129,350								
54	541,080	428,570	340,480	270,000	214,285	169,800	134,325								
56	561,120	444,440	353,090	280,000	222,220	176,100	139,300								
58	581,160	460,315	365,700	290,000	230,155	182,390	144,275								
60	601,200	476,190	378,310	300,000	238,095	188,680	149,250								

distance will give the corresponding current:

$$\frac{4940}{100} = 49.4 \text{ amperes}$$

which is the current required to give 2½ volts drop.

SUMMARY, IN RULES

The methods of using the tables explained in the foregoing may be condensed into rules, as follows:

TABLE 1

TO FIND DROP: *Find in column 7 the number in the line with the size of wire; multiply it by the current and multiply the result by the distance. The final result will be the drop in volts.*

DROP, WITH GREATEST ALLOWABLE LAMP CURRENT: *In column 8 or 9 find the number in the line with the size of wire and multiply it by the distance. The result will be the drop in volts when the wire is carrying the greatest allowable current.*

DROP, WITH GREATEST ALLOWABLE MOTOR CURRENT: *In column 10 or 11 find the number in the line with the size of wire and multiply it by the distance. The result will be the drop when the largest allowable motor is taking full-load current.*

WATTS LOST WITH GREATEST LAMP CURRENT: *In column 12 or 13 find the number in the line with the size of wire and multiply it by the distance. The result will be the loss in watts when the wire carries the greatest allowable current.*

WATTS LOST WITH GREATEST MOTOR CURRENT: *In column 14 or 15 find the number in the line with the size of wire and multiply it by the distance. The result will be the loss in watts when the largest allowable motor is taking full-load current.*

TO FIND WIRE SIZE: *Multiply the current by the distance and divide the desired drop by the product; find the nearest number to the result in column 7; this number will be in the line with the required wire size. Compare this with the insurance requirement.*

WIRE SIZE FOR GREATEST ALLOWABLE LOSS: *Divide the desired loss by the distance and find the nearest larger number to the result in column 12 or 13 for lamps or column 14 or 15 for a motor; this number will be in the line with the required wire size.*

TABLE 2

TO FIND DROP: *Multiply current by distance and find the nearest number to the product in the column under the size of wire. In line with this number will be the drop in volts. If the amperes × feet are greater than the nearest number in the table, the actual drop will be greater than the drop stated in column 1.*

TO FIND WIRE SIZE: *Multiply current by distance and find, in the line with the desired drop, the number nearest to the product. At the head of the column will be the size of wire to use. If the actual amperes × feet are greater than the number in the table, the actual drop will be greater than the drop in column 1.*

If any reader should find difficulty in applying these tables to his own work, a letter to the author describing the conditions and the difficulty will receive prompt attention.

Some Data on Central and Private Power Plant Operation

In a paper recently read before the National Gas Engine Association at Cleveland, Rossiter Holbrook presented some statistics which are of much pertinence to the question of central-station service vs. isolated-plant operation.

The first group of figures was taken from the New York Edison Co.'s semi-annual report to the Public Service Commission for the second half of 1909. According to Mr. Holbrook, this report shows the following significant data:

Total output	120,000,000 kw.-hrs.
Total income	$7,231,602.62
Total station expenses.....	$1,318,570.21
Distribution expenses	1,210,108.84
General expenses	704,400.71
Taxes	402,943.21
Amortization	916,024.11
Uncollectible bills	92,860.08
Total expenses	$4,644,907.16
Total expense per kw.-hr....	3.87c.
Station expense per kw.-hr...	1.1 c.
Average income per kw.-hr..	6.02c.

From these figures it appears that all consumers paying less than 3⅞c. per kilowatt-hour are supplied at a loss which must be made up by charging small consumers extra-high rates. Mr. Holbrook also pointed out that the rate of amortization charged against the company's equipment is about 2 per cent., whereas in discussing isolated-plant operation, central-station representatives commonly figure 10 to 15 per cent.

Referring to the question of whether operating an isolated plant is profitable or not, Mr. Holbrook said:

I should very much like to know why the Commonwealth Edison Co. is operating the plant in the Majestic Theater Building, on Monroe St., Chicago. The company bought this plant at a rather fancy figure, and if the managers had believed in their own talk, it would have been "junked"; but the company is operating it. The same condition exists at Gunther's, the Fisher Building, and the Chicago Title & Trust Building. In all these cases the landlord and tenants are getting bills from the Edison Co., which in turn is operating the landlords' own plant and making money on it.

The New York Edison has installed a little isolated plant at its own coal yards instead of using current from its mains. If the central station will do these things—well, I will leave the rest to your good sense.

I have reports from all over the country, such as the Great Northern, of Chicago, where the total operating expense for 1907 was less than $34,000, as against more than $64,000 for 1905, when central-station service was used; the Missouri Baptist Sanitarium, of St. Louis, that saves $1000 a year out of a total operating expense of less than $5000; the Sohmer Building (an office building), of New York, 29x100, 12 stories, with a yearly cost of $2761.25 for heat, light and power, including 120,000 kw.-hr. of electricity delivered from the plant. The Langsdorf Building, a loft building where the total operating cost for 1910, for light, heat and power amounted to $7757.39, and electricity alone from the Edison company would cost $8198.85. And I have here a report made by Mailloux & Knox on the plant in R. H. Macy & Co.'s store, installed in 1902. They have covered seven years—1903 to 1910—and the cost of electricity is 1.52c. per kilowatt-hour.

LETTERS

Demagnetizing a Watch

In the issue for Feb. 27, S. G. Rose asks for a method of demagnetizing watches. The alternating-current solenoid he describes is a very good method provided one has the current at hand.

A much simpler way of doing the job, however, is to suspend the watch by a strong cord; twist the cord so that when released the watch will spin for several seconds, release it, and while the watch is spinning slowly approach it with the positive pole of any strong magnet and then slowly withdraw the magnet. Do this a number of times and be sure that the magnet is not near the watch when it ceases to rotate or afterward. This method never fails.

C. C. HEYDER.

Hansford, W. Va.

An easy method of demagnetizing a watch is as follows:

Suspend the watch on a string about 2 ft. long and twist the string very tight so the watch will spin for some time when released. Then, holding the upper end of the string high enough to let the watch hang free, release the watch and allow the string to untwist and set the watch to whirling. As the watch whirls, walk toward the polepiece of a dynamo field magnet, and back away again. Doing this two or three times while the watch spins around usually cures the trouble.

THOMAS WILLIAMS.

Gypsy, W. Va.

[Exactly the same suggestion has been received also from W. T. Reeves, Adamston, W. Va.—EDITOR.]

A 1000-ft. ship will be built by Harland & Wolff, Belfast, for the White Star Co. The vessel will be driven at 18 knots by a combination of reciprocating engines and turbines, and her breadth will be 100 ft.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Reversing Gas and Oil Engines

By A. M. Levin

Although Americans hear only occasionally about reversing gas engines, Europeans show a decided partiality to them. The reversible marine gas motor, they say, eliminates the expense of a costly reversing gear, and by its adoption an outfit as light as possible will be obtained. The manipulation in reversing the engine consists simply in the throwing over of one or more reversing levers; excepting that the two-stroke-cycle engine requires generally, as a preliminary operation, to be unclutched from the propeller shaft.

A feature of the two-ported two-stroke-cycle engine in which the scavenging air is compressed in the crank case is that the direction of its motion is optional. The engine will, as is well known, run in whichever direction it is started. To reverse it when in operation, therefore, would seem a very simple matter, as a heavy charge preignited during the compression stroke will tend to do so. A number of patents relating to reversing devices based on this circumstance are on record; many, however, are of rather questionable worth.

Reversing by Preignition

In order that the desired result shall be obtained, it is necessary that the preignition pressure shall become high enough to overcome the inertia of the moving parts during the compression stroke, and, in addition, provide sufficient force to rotate the parts in the opposite direction until a new impulse can be obtained. These two requirements involve a series of others that, primarily, determine the conditions under which the reversal of the engine may actually be accomplished.

For instance, the energy stored in the flywheel, when revolving at full speed, is often several times that generated by a single explosion and, consequently, a first requisite will be to reduce the speed of the engine to a proper limit preliminary to reversal. What that limit should be, as well as what other requirements there are to be considered, will best be revealed by examining the reversing diagram.

In Fig. 1 is shown, in heavy lines, a regular full-load diagram of a two-stroke engine. Assume that, for the purpose of reversing the engine, the ignition be advanced so that preignition of the charge occurs during the compression stroke at the point *E*. The resulting explosion will carry the pressure along line *EF* to a point somewhat higher than the regular expansion line; due, essentially, to the scavenging effect created by some previous strokes during which the explosions have been cut out for the purpose of reducing the speed. From *F*, the charge becomes compressed along the line *FL* to a point *G*. The extent of this compression must evidently be such that the area *EFGHIE* shall represent the energy which the moving parts have transmitted to the charge in compressing it, counted from the time the explosion occurred. The available energy for compression of the gases being spent at the point *G*, the gases will from there

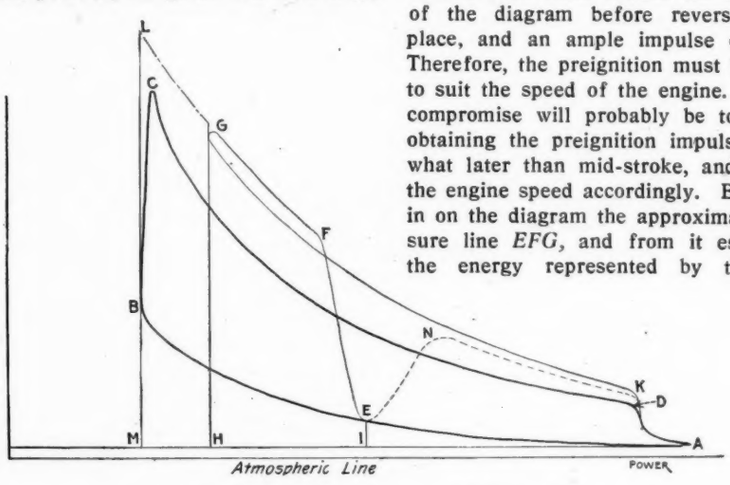


Fig. 1. Diagram of Reversal by Preignition

reëxpand; and in driving the piston backward, reverse the motion of the engine. The new expansion line, which will drop somewhat below the compression line *FG* due to the cooling of the charge, is represented by the line *GK*.

If the energy stored in the moving parts at *F*, immediately after the preignition takes place, is in excess of that represented by the area *EFLMIE*, then the crank will be carried over the center, instead of reversing, and the engine will speed up in the forward direction; the full energy represented by the whole area *LMAKL* contributing to give it velocity. Whenever, at an attempted reversal of the engine, this occurs it will, of course, be needless to make a second attempt without first slowing down the engine. If, as assumed, *G* is the point at which reversal of the moving parts takes place, then the area *GHAKG* represents the work available for driving the engine backward. This work must, therefore, exceed by a proper margin the combined resistances existing or occurring during the first backward explosion stroke, during the compression stroke, and perhaps during part of the succeeding explosion stroke, until a new impulse can be obtained.

As the area *GHAKG* will be larger the later the preignition is timed in the compression stroke, with a slow engine speed late preignition is to be considered favorable for sure reversal of the engine. On the other hand, assuming the preignition to be timed early in the compression stroke and the engine speed high, then the compression might readily be carried back close toward the head end of the diagram before reversal takes place, and an ample impulse obtained. Therefore, the preignition must be timed to suit the speed of the engine. A safe compromise will probably be to aim at obtaining the preignition impulse somewhat later than mid-stroke, and to suit the engine speed accordingly. By laying in on the diagram the approximate pressure line *EFG*, and from it estimating the energy represented by the area *EFLMIE*, the maximum allowable engine speed may be approximated, provided the weights of the principal moving engine parts are known.

It may, of course, be necessary to determine, finally, the most effective speed and ignition point by trial, as different engines act somewhat differently, but this will be a simple matter when the conditions underlying the problem have been observed.

In case the ignition is not shifted quickly enough after reversal to time the following explosion near the end of the compression stroke (which in practice is not readily done by hand), then the impulse will be weak. This impulse might occur even as late as at the point *E*, giving for that stroke the small energy represented by the area *EIANE*. The requirements for a sure reversal in this

case will be that the work represented by the combined areas *GHAKG* and *EIANE* must be ample for keeping the engine in motion through practically four strokes.

It is evident, finally, that the smaller the load carried by the engine when being reversed, the more surely it will respond. It is usual, therefore, to unclutch the load before reversal.

With the stated requirements for a prompt reversal of the two-stroke spark-ignited engine in view, it becomes a simple matter to outline the main features that must be embodied in a reversing device which, with one throw of a hand lever, shall automatically reverse the engine. For instance: The throwing over of the lever may be made to effect the closing of the fuel valve and the clutching in of a suitable controller which, when the speed of the engine falls low enough, will open the fuel valve and advance the ignition to the required point in the compression stroke, and, finally, when reversal has taken place, to shift the ignition back to the ordinary running point. The unclutching of the load, of course, must be done as a preliminary operation.

The Bolinders Gear

The features of a reversing device for the two-stroke-cycle oil engine of the hot-chamber fuel-atomizing type would, in principle, be similar to the arrangement just outlined. An example is the Bolinders reversible kerosene marine engine, the reversing features of which are illustrated diagrammatically in Fig. 2. This engine is fitted with two fuel injection pumps, one, *A*, for continuous work ahead or astern, is timed for firing shortly before the head-end center; the other, *B*, for supplying the fuel to produce only the reversing explosion, which is accomplished at the proper point in the compression stroke. The charging strokes of both pumps are regulated on the hit-and-miss principle by an inertia governor. On the rocking bell crank *C*, which is oscillated by the eccentric *D*, are fastened the two small inertia governor weights *E* and *F*, and the cam blocks *G* and *H* are bolted to the sector *K*, which is fulcrumed as shown and free to swing through a small angle.

During uninterrupted work ahead or astern the regular plunger *A* is actuated by the pickblade on the governor weight *E* in the usual way. At excessive speeds the governor weight will hit the shoulder *L* hard and be thrown up out of line with the nose of the pump plunger, thereby causing the fuel charge to be cut out. To reverse the engine from the ahead motion indicated by the arrow in the figure, the reverse lever *M* is thrown over to the astern position, *N*, thereby compressing the coil spring *O*, which presses the arm *P* of the fork on the end of the link *Q* against the shaft. Friction between the shaft and the fork raises the latter and the link *Q* tilts the sector *K* so as to push the pickblade on the weight *E* out of line with the plunger nose of the pump *A*, cutting out this pump; at the same time, the block *H* is moved to the left, allowing the pickblade *F* to be swung by its spring into alignment with the plunger nose of the reversing pump.

As yet, however, the engine speed is too high and the motion of the governor weight *F* too swift to allow the pickblade to act upon the fuel-pump plunger, but as soon as the speed slows down to the proper point, the governor weight will slide easily up on the cam shoulder *S* and the pickblade will engage the nose of the reversing-pump plunger. The opposing charge for the reversal of the engine is thereby injected and fired at the proper point in the compression stroke. The moment the shaft begins to reverse, the sector *K* will drop to its lower position, putting the reversing fuel pump *B* out of service and the regular pump *A* into action. As the engine requires only a few turns for speeding

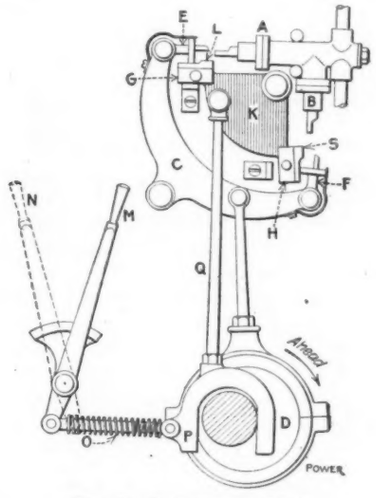


Fig. 2. Bolinders Gear

up, the clutching in of the propeller shaft is done immediately after reversal. The preignition impulse for reversal is somewhat higher than the usual maximum working pressure; a liberal allowance is about 30 per cent. higher. However, as the engine must be built with an allowance for equally heavy casual excess strains, whether it is intended for reversal or not, its weight would not necessarily be materially increased by building it for reversal in the manner described.

Four-stroke Engines

With the four-stroke-cycle engine, the reversing problem becomes somewhat more complicated, because of the necessity that the timing of the valve motion with respect to the crank must be altered. The sequence of the different strokes of the cycle will, of course, be the same whether the engine runs forward or backward; but as plotted on the cam face it will, relative to the one or the other direction of rotation, run in reverse order, due to the reversed rotation of the cam. If, therefore, as may be the most convenient, the compression stroke of the forward rotation be changed into the expansion stroke of the reverse, then the interchange between the strokes becomes:

Forward motion	Reverse motion
Suction	Exhaust
Compression	Expansion
Expansion	Compression
Exhaust	Suction

Normally in carefully adjusted cam motions for four-stroke engines the angles between the center lines of the cams and those of the cam rollers are approximately as follows: With the crank on dead center and ready for the suction stroke, the angle between the center line of the inlet cam throw and the center line through its roller is about 48 deg., and the angle between the center line of the exhaust cam throw and the center line through its roller about 52 deg., the difference between the angles being due to the difference in the most efficient length of the two cam throws. In shifting the cams for reversed motion of the engine, therefore, the inlet cam should be moved ahead 96 deg. and the exhaust cam moved back 104 deg., both counted in the old direction of motion.

Not infrequently, a compromise is made between the differences in the most efficient length of the cam throws, and the same cam is sometimes made to do for both the inlet and the exhaust. In such cases, in order to reverse the motion, both cams should be shifted approximately 100 deg. (for small engines the rule is sometimes to shift them both 90 deg.), the inlet cam being shifted ahead and the exhaust cam backward relative to the previous direction of rotation.

As it would entail some difficulty, however, to shift the cams on the camshaft, this is never done in practice. A simpler way is to mount a second cam on the shaft at the required angle from the regular cam and alongside of it. In reversing, the camshaft is slid bodily lengthwise so as to bring the reverse cam in alignment with the cam roller.

Another method for reversing the timing of the valve openings relative to the crank motion would be to reverse the gear connection between the crankshaft and the camshaft so that the cams always run in the same direction, whether the crankshaft is reversed or not. By this method only one set of cams becomes necessary.

The power for the reversal of the four-stroke engine is always obtained by means of compressed air. In connection with practically all combustion engines of larger power there is required, for starting purposes, an air compressor and

a tank, and the same air supply is used for reversing. The air pressure used is variously of from 120 to 200 lb., but for the sake of better economy, a reducing valve should be inserted in the pipe line between the engine and the air tank. It is imperative for the reliable starting and reversing of the engine that no dead points shall occur during the operation with compressed air, and a sufficient

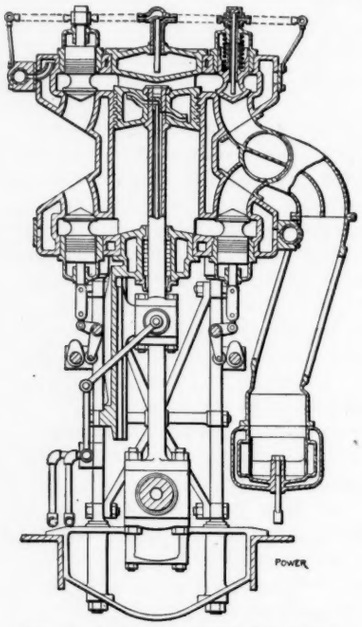


FIG. 3. A "STANDARD" CYLINDER

number of cylinders should therefore be worked by air. The cylinders intended for operation with air are fitted with special air-admission valves actuated by separate starting cams.

In starting or reversing a gas or gasoline engine it is customary, for the more ready handling of the engine by air, to use a reduced compression (about one-half of the regular) in the cylinders operating by combustion. The exhaust valves for these cylinders must, accordingly, be actuated while starting or reversing by special cams that give reduced compression.

THE "STANDARD" MARINE ENGINE

The vertical section through one cylinder of the Standard double-acting six-cylinder marine engine is shown in Fig. 3. The reversing of this engine is effected by means of compressed air which is admitted only beneath the pistons of the three rear cylinders. For the sake of providing against dead points when operating on air, the air-inlet and lower exhaust valves of these cylinders are opened by means of double cams once every revolution of the crankshaft. The opening of all valves is effected by means of a double set of cams; in reversing, the entire camshaft is shifted axially. A small air cylinder and piston are employed for shifting the camshaft and holding it in each position against displacement.

In Fig. 4 is shown the construction of the cams operating the exhaust valves of the upper ends of the cylinders. Both cams and rollers are leveled on their sides, to allow the rollers to slide readily

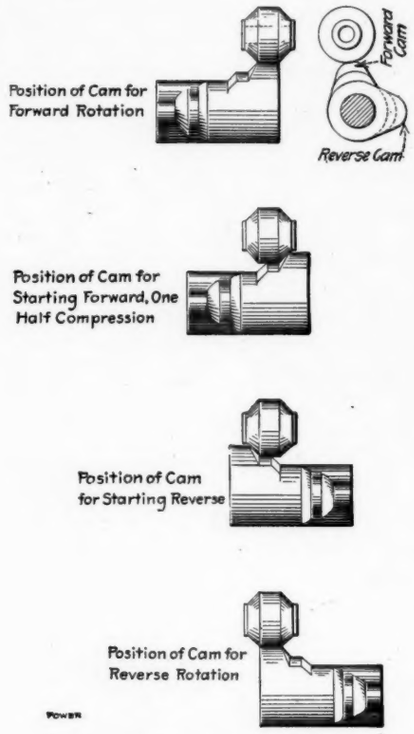

Position of Cam for Forward Rotation
Forward Cam
Reverse Cam
Position of Cam for Starting Forward, One Half Compression
Position of Cam for Starting Reverse
Position of Cam for Reverse Rotation

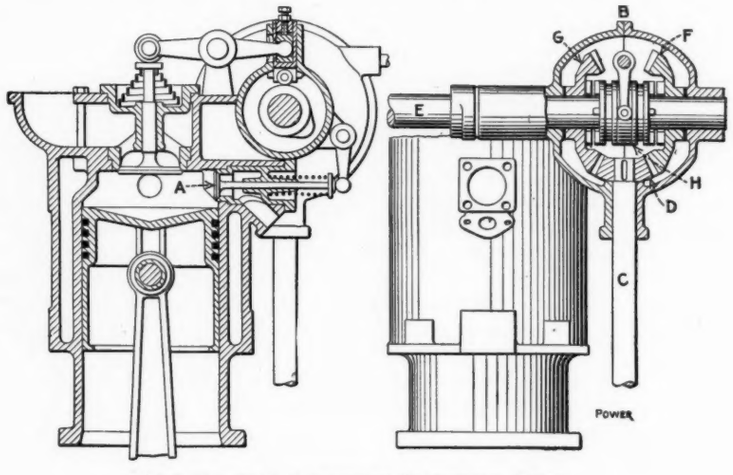
FIG. 4. POSITIONS OF SLIDING CAM

up on the cam throws when the shaft is shifted axially. The cams operating the inlet and lower exhaust valves are of similar construction, somewhat modified to suit the conditions under which they are to work.

For submarine work, the famous Thornycroft firm builds an eight-cylinder single-acting engine the reversing features of which are illustrated in Figs. 5 and 6. Compressed air for starting and reversing is admitted to four of the single-acting cylinders through air valve *A*. There is only one set of valve-operating cams, which rotate always in the same direction, whether the rotation of the crankshaft be forward or reverse. To make this practicable the reversing camshaft gearing *B* is used. The vertical shaft *C* and bevel gear *D* transmit motion from the crankshaft to the camshaft *E*, through gears *F* or *G*. The latter two gears, which are loose on the camshaft, are always in mesh with the driving gear and one or the other is connected to the camshaft by a sliding jaw coupling, *H*, which drives the camshaft by means of a feather and is moved along the shaft by a hand lever.

A SWEDISH REVERSING GEAR

Another reversing gear, which has been adopted by the Sodra Varfs works at Stockholm for relatively small four-cylinder four-stroke-cycle kerosene engines, is illustrated in Fig. 7. Fuel-injection pumps are used, driven by cams on the camshaft. The engine is reversed by means of compressed air which is admitted to all four cylinders in turn, and there are four cams for each of the cylinders, namely, one for the inlet valve, one for the exhaust valve, one for the air valve and one for the fuel admission. These cams are arranged side by side on the shaft *S*. The push-rod which actuates one of the valves is represented at *P* and the cam roller at *R*. The cam roller is not, as usual, connected directly to the valve stem or pump plunger, but mounted on a lever *L*, through which it transmits motion from the cam to the stem or push-rod. For disengaging, in maneuvering, any of the valves or pumps, there is used a control shaft *C*, which carries cams in line with the various cams on the camshaft. If the shaft *C* be turned to the position where none of its cams touch the toes *G* of the levers



FIGS. 5 AND 6. CAMSHAFT REVERSING GEAR

L, then all the stems become inoperative, because the coil springs *Q* will hold the rockers *L* in the position shown by the dotted lines, allowing the main cams to clear the cam rollers *R*. Of the cams

shown on the shaft *C*, *A* belongs to the compressed-air valve, *F* to the fuel-injection pump, and *V* to the inlet and exhaust valves.

In reversing, the maneuvering shaft *C* is first rotated by means of a handwheel so that the point *O* is opposite the toe *G*, and the camshaft *S* is then moved axially so as to bring the reverse cams for the suction and exhaust valves into working position. The control shaft *C* is then rotated further so that the cam *A* comes opposite the toe *G*, in which position the compressed-air valve corresponding to the cam *A* and the inlet and exhaust valves corresponding to the cam *V* will be operative. The air pressure thus admitted will first check the forward motion of the engine and then give it speed in the reverse direction. The shaft *C* is thereupon rotated so that the cam *F* is opposite the toe *G*, causing the compressed air to be cut off and the fuel pump to be put in action.

The Benz Reversing Engine

The Diesel marine engines built by Sulzer Brothers, and those built by the Benz firm at Mannheim, work on the two-stroke cycle, and they are equipped for scavenging purposes with large air pumps driven by cranks on the main crankshaft. The main features of these engines are generally similar to all two-stroke engines of the Diesel type, but the two methods adopted for their reversal may well be worth outlining.

In the Benz engine there are two double-acting scavenging air pumps driven by cranks set at 90 deg. A central extension of the piston of one of these pumps constitutes a compressor plunger for supplying pressure air for reversing and starting and an extension of the other piston forms a second-stage plunger for compressing high-pressure air for fuel injection. The air for reversing and starting purposes is stored at a pressure of 200 lb. in a receiver. The underlying idea of the reversing arrangement is to transform the large pump cylinders, which under ordinary operation supply the scavenging air, into air-motor cylinders or of the reversing of the engine.

The pumps draw, ordinarily, air from the outside and deliver it at a low pressure into a holder which connects and surrounds the air ports of all the combustion chambers. When each piston, at the bottom of the stroke, uncovers these ports, which are cut in the usual way in the lower part of each cylinder, scavenging air is admitted. In maneuvering for reversal, however, the air valves controlling the intake to the pump cylinders are reversed, so that instead of the pistons drawing air from the outside they will receive air from the compressed-air receiver, reduced to about 60 lb., and discharge to the atmosphere, or to the scavenging-air holder. That is, the pump cylinders become changed into air-motor cylinders which resist the forward motion of the engine and finally reverse it.

The scavenging air supply to the combustion chambers is, in the meantime, not cut off, because the discharge from the air cylinders will, in part, be supplied for scavenging. The combustion

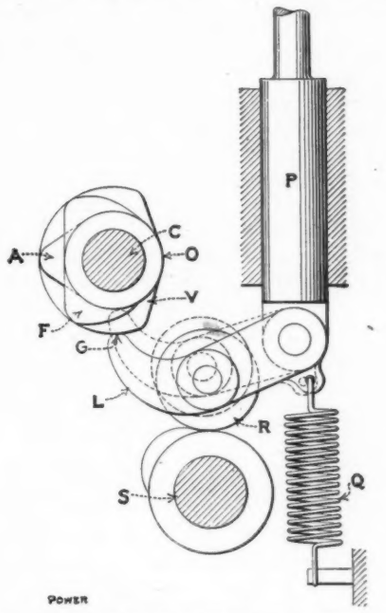


Fig. 7. A Swedish Reversing Gear

cylinders are, therefore, always under normal working conditions and ready to act in harmony with the air cylinders. In reversing, the fuel is, of course, cut off for a short period to allow the engine to slow down, and the timing of the fuel-injection and exhaust valves reversed. The latter is done by the axial shifting of the camshaft, which is fitted with a double set of cams.

Sulzer's Method

For the Sulzer engine the reversing is done by means of compressed air admitted through the regular starting valves directly into all of the working cylinders. A tandem two-stage air compressor, connected to a crank on the main shaft, just forward of the scavenging air pump, supplies the air pressure required for fuel injection and also for starting, the first stage of the compressor delivering to a receiver in which the pressure is kept at about 180 lb., and from which the air for starting and reversing is obtained. The second stage, compressing the air to a very high pressure, supplies the tubes from which air for the fuel injection is drawn.

From the foregoing it is evident that several practical methods are available for making the combustion engine reversible, any one of which, if rightly carried out, ought, especially in heavy marine work, to be preferable to the use of reversing gears or adjustable propeller blades, which at present are the more common methods in use for vessels of small and moderate sizes.

LETTERS

Cost of Fuel for Diesel Engines

In the Mar. 5 issue is an article by R. E. Mathot, entitled "Comparison of Gas, Oil and Steam Motive Powers," in which the Diesel engine is put at a serious disadvantage by reason of the high price of fuel which the author assumed. The price of $1.40 per kilogram, making the operating cost 0.028c. per brake horsepower, might easily be misleading to readers who are not acquainted with the fact that the figures apply only to European conditions—which we presume must be the author's intention.

Fuel oil such as is used in the Diesel engine costs in this country on an average less than one-half of the price stated by Mr. Mathot. Basing the calculations on the consumption of fuel of the respective prime movers as specified in the article, and applying to them actual cost of the respective fuels in this country, will easily show that the Diesel engine offers the lowest operating cost.

Busch-Sulzer Bros.-Diesel Engine Co.,
By A. O. Krieger.

St. Louis, Mo.

[All of Mr. Mathot's figures were based on European practice and conditions and therefore do not give comparisons that are obtainable in the United States. As Mr. Mathot is a well known Belgian engineer, we presumed that these facts would be recognized by our readers. Mr. Krieger's comment on the price of fuel oil is quite warranted.—Editor.]

Pittsburg Greatest Manufacturing District

The total primary horsepower which drives the wheels of Pittsburg, Penn., shops and mills during 1909 was 791,047, according to the industrial census just compiled by Director E. Dana Durand, of the United States Census Bureau, and announced by the Pittsburg Industrial Development Commission.

The figures represent the metropolitan district, an area within a radius of 15 miles of the center of the city embracing many populous communities outside the municipal boundaries. Several of the largest manufacturing establishments are in these suburbs and the statistics of the industrial census make this center the greatest manufacturing district in the United States.

This aggregate horsepower, which is practically sufficient to drive all the big vessels in the navy, is generated in 200 different classes of industries.

The greatest power is generated in the iron and steel works and rolling mills where 54 establishments used 357,945 hp. in 1909, the latest year for which statistics are available in the Federal Census Bureau.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Dust in Relation to Heating*

By Conrad Meier

It is a popular notion that steam and hot-water systems give a different sort of heat. Engineers would define this more correctly by stating that various methods of heating will produce different air conditions. The old-fashioned devices are still considered by many people to be more wholesome, even though less efficient than the modern apparatus, of which it is often said that it does not ventilate, the dead air and stuffiness being attributed generally to the lack of air supply incidental to heating.

Partly out of this idea has grown the popularity of the indirect system, but now complaints are heard, that through heating, and even tempering, by stacks, the air will lose its natural sweetness and refreshing qualities. The recent tendency by the medical profession to oppose plenum ventilation appears to be the result of such observations. Engineers themselves will have observed, especially in public buildings and schools, that even a generous plenum supply will not make the occupants forget the desire to open the windows.

Heated but unoccupied rooms are often found stuffy, even after long vacancy, when natural ventilation, which is always present, should suffice to keep them sweet. Everybody has noticed the stifling air of ill kept, empty railroad coaches which is not altogether caused by overheating, nor by previous occupancy. In such cases the foulness of the air cannot be due to lack of ventilation alone; these circumstances indicate a cause of vitiation aside from the well known sources and explain in a measure the demand for purer air which has really arisen and become general only with the introduction of the modern ways of heating.

None of this evidence is proof that the heating apparatus itself is at fault, but it should certainly raise the question whether the quality of room air is altered in any material way by the process of heating.

Causes of Vitiation

As proof of certain chemical changes there are the investigations of Profs. Esmarch and Nussbaum. The former authority has demonstrated the following points:

Dust, especially when of organic nature, will decompose on heating surfaces from a temperature of about 160 deg. F. upward. Of the gases produced, ammonia is shown by chemical reaction not only over the radiator but in the room air at large. The quantities are usually too small to be called a poison to the average person, but are quite often noticeable by odor. Under extreme conditions, when steam is turned into a dusty radiator or stack, the ammonia may become very disagreeable. Analysis was made for CO, but none could be found.

When heating surfaces were carefully cleaned before the test, the effect on the air was still present and demonstrable by analysis, but less intense, thus showing that the dust suspended in the air will also decompose when passing over the heating surfaces, and that there is a constant vitiation going on besides the periodical effect while reheating. No traces of any foreign gases could be detected until steam was turned on; hence it is safe to say that heating is the cause. It may be concluded also, that if heating were accomplished by clean surfaces and without causing air circulation, decomposition could not take place.

Prof. Nussbaum, investigating independently, has arrived at the same general conclusions, but has gone further into certain details. He found that moist organic dust will begin to decompose even at 140 deg. F., while dry dust, or dust in dry air, must be heated to about 200 deg. F. before gases become noticeable. The same applies to dust lying on surfaces or passing by them.

Means to Prevent Vitiation by Heating

Naturally, the foremost remedy that suggests itself is to exclude the dust as the main source of vitiation. All will agree that this would be desirable, but unfortunately it cannot always be done. Even where artificial devices for this purpose are used in connection with warm-air heating and air-supply systems more or less dust will enter into a building in other ways. Vacuum cleaners and general vigilance will help, but the fact remains that a great majority of cases will have to contend with more or less dust.

The safest way to prevent the chemical adulteration of air through decomposing of organic matter would appear to be a moderate temperature of heating surfaces. With clean surfaces and dry air, according to Nussbaum, there will be no vitiation with temperatures up to 160 deg. F., and none to speak of until the surfaces are near the boiling point. Under less ideal conditions, when more or less dust is to be expected, the temperature should be kept down to about 160 deg. F. or lower, as a rule, accepting an occasional rise to 180 deg. F. during a cold spell as unavoidable and of minor importance.

Moderate temperature of heating surface seems also advantageous in view of the devitalizing of the air by the absorption of all the ozone and probably of part of the oxygen.

Hot-water heating would seem to meet the situation best in respect to temperature and liberal surfaces, but whichever system is used, it is advantageous to reduce the tendency to create air currents and to give preference to the kind of surface that is yielding a greater ratio of its heat by radiance. Low and flat radiation which is recommended is generally no more expensive considering its higher total efficiency.

The next important rule should be to insist on clean heating surfaces. It will not do to depend for this on the good intentions of anyone concerned. Engineers must take the first step to induce or compel clean surfaces by selection of the most easily cleanable styles and by proper disposition to keep the greatest part of it in view. As it happens, the styles which are giving the best radiant heat and the least contamination by dust are also the easiest to clean. Radiators should have fair spacing between loops to show the dirt and to make it likely that it will be removed. They should not be tucked away in corners. Better means are always at hand to make them inconspicuous or presentable, as the case may require. Enamel finish in some neutral color, black or white, is only one of those means, which, by the way, will help appreciably in the radiating efficiency. This sort of finish, it is sometimes complained, would show the dust. It is all the more desirable then, in order to induce clean surfaces.

Considering that the requirements for heat and for ventilation generally are not all coincident, and that a purer, more effective and wholesome air supply can be secured by windows or other rational means, when and where required, and without affecting the heat supply, it

*Abstract of paper read before the American Society of Heating and Ventilating Engineers.

would seem that the idea of better sanitary conditions through heating by air is largely a delusion. The indirect method should be restricted to cases where it is really indicated, or, if no other system will do, the conditions might be made more favorable for it by extra protection. In any event, such apparatus should be designed to avoid excessive air temperature, equipped with filters to keep the dust out of air passages and heating stacks, and with all provisions to induce proper maintenance of the apparatus.

The excessive dryness complained of sometimes with direct steam heat is often nothing but irritation of the respiratory tract through dust. It should be relieved in such cases by cleaner radiation and cleaner surroundings, and by lower temperature of heating surfaces. Moistening pans will raise humidity, but generally do not improve the air while dust and dirt are present.

These would seem to be sufficient reasons for reform, and the lines on which to improve heating apparatus from the sanitary point of view do not seem difficult to follow and would lead, if anything, toward greater efficiency in other directions. Nor should there be much difference in cost when taken as a whole. Hot-water heating, for instance, can be brought down very nearly to the cost of steam, if scientifically planned, and the practical possibilities of its adaptation to all sorts of conditions are by no means exhausted. The ways and means for reform may therefore be said to be within reach.

Briefly resuming, the following rules may be put down:

1. To prevent chemical vitiation of room air by heating it is desirable to have heating surfaces at moderate temperature, if possible not exceeding 180 deg. F. at any time and to use easily accessible and clean radiation. Air supply should be filtered where necessary.

2. To prevent the physical admixture of dust and its drying, caused by the air currents due to heating, it is desirable again to use heating surfaces at moderate temperature, well distributed, spread out and inducing salubrity in general.

3. To reduce the pollution of the air by the bacteria the same means are indicated as those recommended to keep down the dust.

4. To produce thermal conditions most agreeable and wholesome to a majority, it seems desirable to favor the application of radiant heat which warms the surroundings and occupants directly and produces comfort at lower air temperature. It should be used in a mild form and be well distributed. Ventilation should be treated as a separate problem, as each can be solved by itself to the best advantage.

5. To prevent excessive dryness, the same idea of keeping down air temperature by utilizing heat rays seems indicated.

6. Ventilating apparatus should be made self-cleaning throughout.

Charge for Steam Heat

Referring to C. E. Diamond's question of a proper charge for steam heat, in the Feb. 27 issue, I had occasion to make a charge for heating about two years ago, and made it on the following basis, which is still carried out:

Each square foot of radiating service will require about ¼ lb. of steam at the boiler per hour. Assuming a loss of 10 per cent. on account of condensation, the actual amount of steam required at the boiler will be 0.277 lb. per square foot per hour. Under good conditions 1 lb. of coal will generate 10 lb. of steam, and on this basis there will then be required 0.0277 lb. of coal per square foot of radiating service per hour. From the number of square feet of radiation supplied, the total coal required can then easily be determined and the charges based on the cost of the coal per ton.

The cost of operating the plant to give the above condition will simply be in the same ratio as the amount of coal used for heating to the total amount of coal used in the plant. For example, if the heating service requires 10 per cent. of the total coal burned, the charge would be 10 per cent. of the coal bill plus 10 per cent. of the other operating costs in the plant. There will, of course, be some variation due to extreme cold and mild weather, but the above will give a fair average.

If more exact conditions must prevail, it might be advantageous to install steam meters and base the charges on the percentage of steam used for heating to the total amount of water evaporated in the plant.

W. R. METZ.

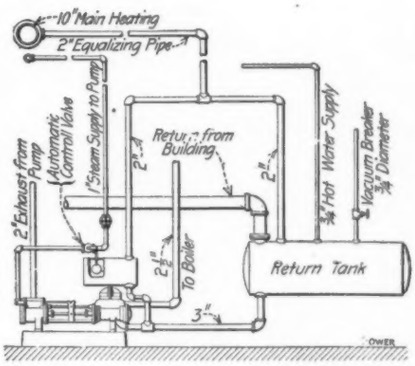
Washington, D. C.

Why Does This Pump Race?

I am employed in a steam-heating plant, the heating being done by three return-tubular boilers, 66 in. by 18 ft., through a two-pipe, direct, dry-return system. The building is seven stories high. The returns empty into a tank 6 ft. long and 2 ft. in diameter, and from there are pumped back to the boiler by a Deane duplex pump, 7½ and 4½ by 12 in. The pump is automatically controlled by the height of the water in the tank; when the tank is half full the pump will stop and as the water rises it will go to work again.

Held up from the floor by a support, to the height of the top of the discharge chamber of the pump, is an equalizing tank. The steam-supply pipe comes down and rests on the top of this tank. At this point there is a steam valve in the pipe that is connected to a rod running down into the equalizing tank. To this rod is attached a ball float, and there is a ball and lever connected to the top of the valve. The height of the water in the equalizing tank corresponds to the height of the water in the return tank; so that any variation of water level will cause the steam valve to open or close. An equalizing pipe from the steam main is connected to and has free opening to both the equalizing and return tank. The water supply to the pump from the return tank is piped to the suction chamber and just outside the chamber this pipe is tapped and a branch run to the bottom of the equalizing tank. The pump discharge is piped direct to the boilers. There is a pipe leading from a hot-water tank to the return tank for the purpose of increasing the supply to be pumped to the boilers. A vacuum breaker, operated by opening or closing a valve, is placed on the top of the return tank.

These boilers carry 80 lb. steam pressure, and the pressure on the heating system is 4 lb.



PIPING TO PUMP AND RETURN TANK

Now when the steam heat is shut off and the vacuum breaker opened, the pump, instead of working along as it has been doing, will suddenly begin to race, starting slowly and speeding up, and if let alone would tear itself to pieces. At the end of the stroke there is a violent noise. I shut the pump down and wait until all escaping steam and air have stopped at the vacuum breaker; then I start the pump and no more trouble is encountered until the next time.

The man who installed this plant says it cannot be run without the equalizing feature, but with this feature cut in or out it makes no difference in the action of the pump. It may be that some of the readers of POWER have had a similar experience and can give an explanation.

W. T. MOULTON.

Portland, Maine.

Belgium produced 23,125,200 tons of coal last year, but it had to import between three and four million tons.

Stereophagus Sewage Pump

This pump, which is of the centrifugal type, has been designed for pumping sewage water. It is manufactured by R. C. Parsons, 39 Victoria St., London, S. W., England.

In Figs. 1, 2, 3 and 4 is illustrated the construction of the interior of the pump and the impeller, which is made conical. The liquid is admitted to the impeller on one side only. There is also but one side or shrouding to the impeller from which the blades project, their edges terminating in a conical shape.

Surrounding the impeller is a spiral chamber leading to the outlet pipe, similar to the ordinary centrifugal pump. Attached to the interior of the inlet pipe, and projecting as far as the center is a fin having its upstream edge sloping gradually to the side of the pipe, as shown at *A*, Fig. 3. The casing on the conical side of the impeller is formed to the same angle as that of the impeller, so that the edges of the blades fit against the internal surface.

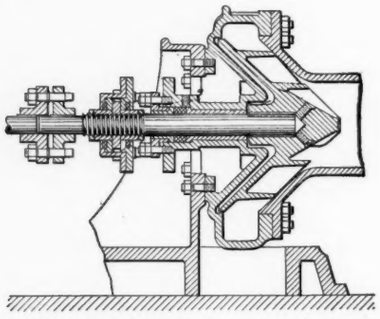


FIG. 1. SECTIONAL VIEW OF PUMP

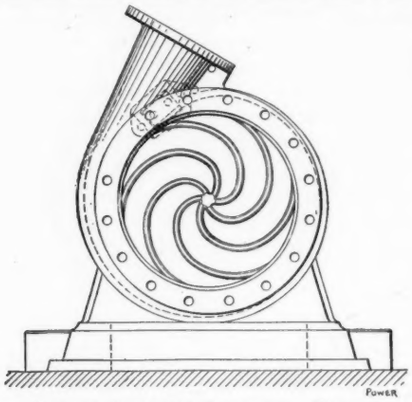


FIG. 2. LOOKING INTO THE CASING

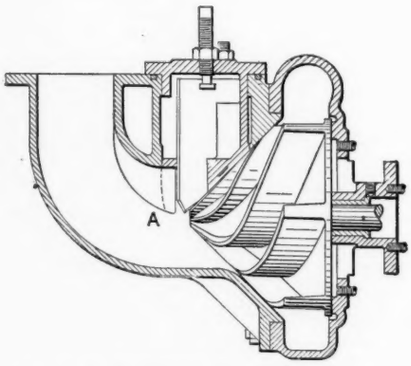


FIG. 3. SHOWING SLICING KNIFE

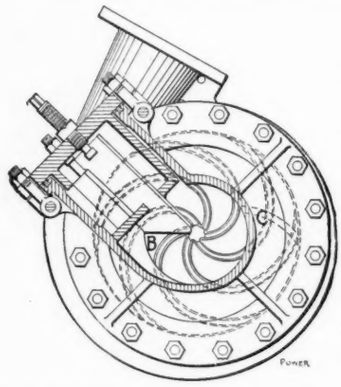


FIG. 4. CASING FROM THE KNIFE SIDE

At the side of this casing is a cylindrical chamber, in which is a holder containing a steel knife with a sloping edge so as to coincide with the angle of the blades of the impeller, as shown at *B*, Fig. 4.

The upstream side of this knife is in line with the fin and its point reaches to the center of the impeller. In the conical surface of the casting is a spiral groove having square edges, as shown at *C*.

The action of the pump is as follows: Having been charged and set in motion, it draws the liquid in through the suction pipe which passes between the blades of the impeller into the spiral chamber and is injected to the delivery pipe. Should there be any solid materials in the liquid, they slide along the fin and, coming in contact with the blades of the impeller, are carried around until they reach the edge of the knife and are cut to pieces. Should they not be cut small enough to pass through the passage between the blades the first time, they are carried around and are recut as often as required to enable them to pass through the impeller.

The passages between the blades of the impeller at the point where the cutting takes place are arranged to be of the least dimensions of any of those in the pump and to enlarge therefrom toward the outlet, so that no piece of solid substance which passes the knife can become fixed in any of the passages.

In the event of any substance becoming jammed between the edges of the blades of the impeller and the casing, they fall into the spiral grooves in the face of the conical casing and are ejected with the fluid that is being pumped. The cutting action of the impeller blades as they pass the knife is at one point at a time, and therefore no perceptible shock occurs when any substance is cut up.

Means have been provided for adjusting the impeller toward the conical casing and as the edges of the blades become worn they are brought closer.

Vulcan Automatic Drain Valve

A device used in conjunction with the Vulcan soot cleaner is an automatic drain valve, as shown in Fig. 1. It can be used for other purposes. G. L. Simonds & Co., 64 East Van Buren St., Chicago, Ill., are the manufacturers.

This valve is to prevent water getting into the piping due to leaks which might occur in the operating valves, discharges any water that might get past a leaky valve and also indicates such leakage.

It is a gravity check valve and is placed just beyond and below the operating valve on the no-pressure side, being set vertically and connected to the main horizontal line overhead by means of a nipple and return bend, as shown in Fig. 2.

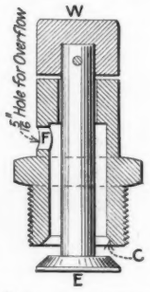


FIG. 1. SECTION THROUGH VALVI

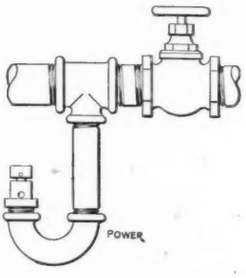


FIG. 2. APPLICATION OF VALVE

The valve is made of brass and consists of an outside casting threaded at the bottom as shown. The bottom of the valve chamber is finished off in the form of a valve seat at *C*. The disk *E* is thrust upward when the steam pressure is turned on, thus making the escape of live steam impossible. The instant the pressure is released, the combined weight of the disk, valve stem, and weight *W* causes the disk *E* to drop, thus leaving a free vent through the opening *F*.

Any steam or condensation leaking through the globe valve in Fig. 2, cannot get into the piping system as the water would drop down the return bend and flow out through the opening in the drain valve.

Recently a collier at the coal pier of the Baltimore & Ohio R.R., in Baltimore, was loaded with 7029 tons of cargo coal and 545 tons of bunker coal in 4 hr. 30 min., thereby establishing a record in coal-handling.

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POWER

Cleanliness in the Power Station

Cleanliness and system are two of the prime requisites that make for satisfactory operation in the electric power station. They cannot be too strongly emphasized, and if they are not rigidly observed, the best installation will fail of its purpose and the money expended on it will, to a great extent, be wasted. Every piece of the equipment should be kept neat and clean and its installation so arranged as to permit of this being easily accomplished. For example, the switchboard and controlling apparatus should be placed out from the wall a distance sufficient to allow the operator to get behind them easily and this space should be kept free from all material not directly connected with the operation of such apparatus. That this is not always done is only too true, as a casual glance behind many a switchboard or machine will testify.

The use of switches and wires as hangers for articles of clothing, etc., is a practice sometimes seen and one that should never be allowed; metal lockers, well ventilated and of sightly appearance, are a necessary adjunct to a properly equipped power house. Individual metal receptacles provided with hinged metal covers or lids for the accommodation of oily waste or waste papers should also be considered necessary parts of the equipment. Such things as these are potent factors in the reduction of fire risks.

The power station should not be used as a storage place for spare parts and for the storage and filtering of oil; where possible, a separate building should be used for such purposes. A power station large enough to contain a miscellaneous lot of material that might be stored in it is too large for the real purpose for which it was built, which usually means that more money was expended on its erection than was necessary. It is considerably cheaper to build storage houses than power houses.

The windows should receive sufficient attention to keep them in efficient condition and if any are located near electrical apparatus they should be so arranged that they can be closed quickly and easily to prevent any damage that might result from rain beating in. It is also well to have all lamps, as far as practicable, so located as to be accessible for trimming or cleaning from the floor, platform, walks, etc. It is usually found that in the best lighted stations the lamps are thus arranged, for the easier it is to care for them, the better attention they will get.

A fire hose conveniently located and an easy connection to a water main have been found a very profitable investment; in addition to this, a few buckets or boxes of dry sand and a fire extinguisher or two will be found highly advantageous in case of fire in a part of the plant where water cannot be applied.

It may be possible that some of the foregoing suggestions will seem quite extravagant and their adoption to entail unnecessary expense and trouble; nevertheless, their strict observance will result in more pleasant surroundings for the men in the station, far less risk of fire, lower insurance rates and less liability to service interruptions.

A clean and well ordered power house is indicative of smooth running, well kept machinery and, consequently, a low operating cost, which is, after all, the main object for which the engineer should strive.

Using Second Hand Machinery

Frequently there are cases where apparatus has been shipped for installation that was defective in some of its parts; perhaps the exterior presented a finished appearance, but some detail of interior construction may have prevented proper operation.

The small boy is expected to take apart almost anything that falls into his hands because he wants to know how it is made,

but it is not advisable for an engineer to disassemble machines which arrive at the plant for installation; he should make sure, so far as it is possible, that they have been properly assembled and are in good condition.

This is especially true in the case of large concerns where the machines in one plant may be discarded for a time and then installed in another for permanent use.

A case has just come to light where a motor-driven centrifugal pump was to be used as a boiler feeder. When the pump arrived the engineer figured that it was capable of delivering water against a pressure of but twenty-three pounds. He was assured, however, that it was good for sixty pounds pressure; and the pump was installed. After it was tested out, the best to be obtained was a delivery pressure of twenty-one pounds. Of course, the work of installing and dismantling was a dead loss, as well as the service of the pump. Either carelessness was responsible for this blunder, or else it was a case of "put it in and see what it will do." Such practice is not advisable in power plants and is usually disadvantageous to all concerned.

A duplex steam pump was next delivered to the plant, the connections were made and the steam was turned on, but the pump refused to move. An examination showed that the pistons were so badly rusted in the cylinders that it was necessary to use a hammer and a block to loosen them, and the pump was badly corroded in the steam chest and other interior parts. The pump was made to run after considerable trouble, but it would not deliver sufficient water.

Another pump was installed and before it had been in operation an hour the piston came off the rod and the pump was shut down for repairs. When it came back from the shop the engineer found that the threads on the plunger piston were of a special size and that a nut one size too large had been used. The locknut had been omitted, and if the pump had been started there probably would have been another accident.

These are little matters, but they go to show that someone must have been careless.

Second-hand machinery may answer in some places, but the power plant is not an ideal place for trying them out. It would doubtless be money well invested if such apparatus were tested out before being delivered for installation; certainly this would avoid useless work and costly delays in many cases.

Prepare for the Summer Now

As summer is approaching, with its hot, sultry days—days when the engineer's labor should be as light as possible—preparation should be made to care for those small things which, individually considered, are of little importance, but which collectively promote comfort and facilitate the operation of the plant.

Perhaps there are leaky flanges and fittings with sand holes in them through which steam hisses the announcement of a remediable loss. The steam headers on top of the boilers with their many branching lines may be the source of erosion and damage to brickwork by reason of the constant dripping of water of condensation. To change over from one header to another in a large plant is not a pleasant task; many valves, usually stiff and difficult to turn, must be opened and closed, and this is much less arduous while the weather is cool than in midsummer; much trouble and inconvenience may be avoided by "doing it now."

In the cellars and tunnels like conditions may prevail. Work in such hot, close quarters may be much more comfortably undertaken at this season, especially as it has to be performed in a short time in order to prevent any interruption of the service during a period when service is needed most.

It may be that the bridge-walls and arches require patching, or sections of the grate bars need renewing and the boilers cleaning. Perhaps the soot pits, breechings and chimney base are full of soot, and thereby seriously interfere with the draft and the boiler economy.

The feed-water filters surely will better perform their function if they are cleaned and the filtering medium is renewed. Warm weather increases the demands on the refrigeration plant, and to meet them the plant must be in good order.

It is very probable that the coal-handling machinery is worn, and the belt housing contains fine, hard-packed coal which prevents the guide pulleys from working freely.

Now is the time to make an effort to systematize work that is so essential to the safe and efficient operation of the plant; much can be done thoroughly and comfortably at this season which if neglected until the hot summer days would be slighted or left undone.

Load Factor

Engineering practice seems to be handicapped in many respects by a lack of definite standards by which the performance of similar installations and apparatus may be readily compared. Important among these is "load factor," which term is interpreted in so many ways that much unnecessary confusion results.

A prominent engineer (presumably an expert in his line) was asked upon the witness stand recently to define what is meant by "load factor." He replied that any one of several answers might be made, each equally correct according to the interpretation of the conditions. He stated further that he had a conception of what should properly be designated as the load factor, but any other engineer holding a different opinion would be equally justified in the light of present practice.

This witness summed up the situation very clearly and concisely. By many the term "load factor" is used to mean the ratio of the total yearly output in kilowatt-hours or horsepower-hours to the rated yearly capacity in similar units. Others regard it as the ratio of the output to the maximum continuous capacity; while still another group of engineers, notably in English practice, consider the load factor as the ratio of the average load to the maximum load carried at any time during the year.

The first of these methods might seem a rational basis provided all similar machines were rated according to the same standard, but unfortunately the practices of manufacturers also differ in this respect. However, we shall not attempt to go into the merits or demerits of any of the methods mentioned, but merely wish to call attention to the need of some definite basis being selected and observed. Since the cost per kilowatt-hour is directly dependent upon the load factor, the selling price is often similarly determined; hence the importance of formulating a definite basis.

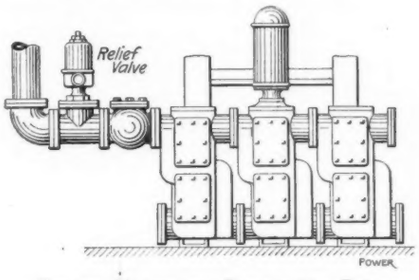
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Mine Pump Gasket Blew Out

An experience I once had with a 6x8-in. triplex power pump, driven by a 50-hp. motor, and located at the bottom of a 500-ft. shaft, may be of interest to mine engineers.

To place the pump it was necessary to take it to pieces, and reassemble it in the mine. In so doing, the bolts that hold the cylinders to the base probably were not drawn up evenly, for on starting up the gasket blew out. A new gasket was cut from common red sheet packing, and the pump started, but this gasket lasted only a few minutes.



RELIEF VALVE ON DISCHARGE PIPE

The sump was filling fast and something had to be done quickly. As I was out in the country I had to use what material I had on hand. A new gasket was cut from the same sheet packing; a wire fly screen was taken from a window and a gasket cut from that and placed on the top of the sheet rubber, and the pump started, but not before the water was up to my knees. That gasket has been in use for six months and is still tight, although another one ordered from the pump factory is in readiness if needed.

This pump is at the bottom of a 500-ft. shaft and the discharge pipe has no stop valve. A gage at the pump registers 230 lb. per square inch. Twice the pressure has burst the tube in a 300-lb. gage and once cracked an air chamber. As there can be no pressure in excess of the head, I was at a loss to account for what was very evidently an increase in pressure at times. I finally decided that it was due to water-hammer caused by the pump being allowed to run after the sump was empty.

As the pump sucked air, and the water surged, in the vertical discharge pipe, I put a check and a relief valve in the pipe as shown, and have had no trouble since.

E. J. SAXE.

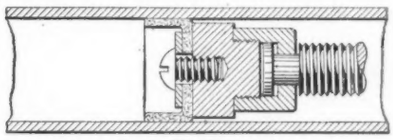
Christopher, Ill.

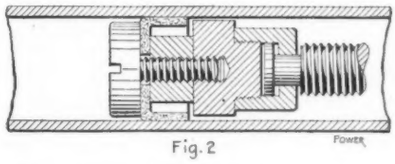
Calibrating Gages

Most progressive engineers have the conventional screw force pump for comparing pressure gages. This pump, with slight modification, is also useful for comparing vacuum gages. The one I use has a bore of 1⅛ in. and the plunger is arranged as shown in Fig. 1 for testing pressure gages.

It is also an excellent and simple device that can be easily reversed by turning the leather around, fitting a new screw and washer to look as shown in Fig. 2.

Care must be taken in filling the pump with water to have the pipes full and the plunger just covered. After a few trials one will become expert in obtaining 22 in. or more of vacuum. Then correct the faulty vacuum gage by resetting the index.





SECTIONAL VIEWS OF PRESSURE-PUMP PISTONS

FRANK MARTIN.

College Point, N. Y.

Cleaning Air Compressor Cylinders

For cleaning air-compressor cylinders I tap a ½-in. hole in the suction line near the cylinder and screw in ½-in. pipe. Running vertically from this is a 2-in. pipe, 18 in. long; on top of it I put a screw cap and on the ½-in. line I put a globe valve.

Once a week I use 2 gal. of hot water in which has been sliced a cake of soap. I stir it thoroughly until all of the soap dissolves; then gradually pour in the top of the pipe, through a screened funnel, using about a cupful at a time. After using about three-quarters of the suds, I pour in one-half a cup of oil, and wash out with the balance of the suds. The oil is then put in at the suction inlet, so as to lubricate both upper and lower parts of the cylinder; when oiling with lubricators the oil does not get to the bottom of the cylinders.

Where there are two cylinders, of course the piping is also attached to the low-pressure one.

A. P. SHAW.

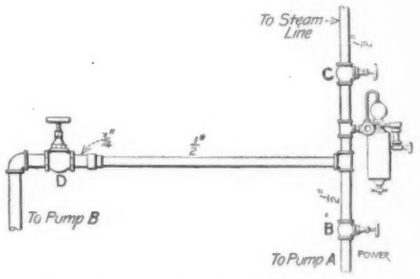
Lima, Ohio.

Why Won't the Lubricator Work?

I have a lubricator puzzle which I would like to have some reader of POWER solve for me.

I installed an extra boiler-feed pump a short time ago and, not wishing to buy an extra lubricator, I piped the steam so that one lubricator and throttle valve answered for both pumps. I placed a valve *B* and *D* in each pipe between the pumps and throttle valve *C*, as shown.

The pump *A* is used regularly and after starting it up I found that the lubricator would not operate and, thinking it might be stopped up, removed it and gave it a thorough cleaning. It balked just the same, so I put on another and it acted in the same way. As a last resort, I removed the valve *B* and the lubricator works satisfactorily. I tried to operate the lubricator with the valve *C* wide open and using the valve *B* as a throttle valve, but the lubricator acted the same.



LUBRICATOR AND PIPE CONNECTIONS

When using the pump *B* with the ¾-in. valve *D* open wide and using *C* as a throttle, the lubricator works all right.

Now, why will it not work with the valve *B* open and using the valve *C* as a throttle?

The pump *A* has a detachable valve rod, so I can lift it off and stop the pump when I wish to run the pump *B*. I am going to put a ½-in. stop cock where I had the valve *B* and see if it will help any, as I want a valve at that point.

W. S. HULL.

Sheldon, Ill.

Riveting on Saddle Flanges

Owing to the installation of a new engine and condenser, it was necessary to make some changes in the discharge line which served two other condensers. These could only be shut down for about two hours each day. Owing to the elevation of the discharge line, it stood full of water when not in use; hence it would have to be drained and dammed up each night.

As one of the jobs was putting a 30-in. saddle flange on the under side of the 36-in. end of the discharge line, to get a siphon effect, the job would require considerable time, as the bolt-holes would have to be drilled, the 30-in. piece drilled out and several 30-in. fittings placed before the valve was put on.

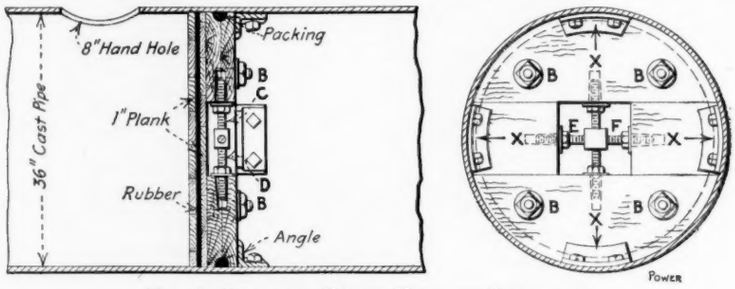


FIG. 1. SIDE AND FRONT VIEW OF BULKHEAD

The bulkhead shown in Fig. 1 was used, which is almost self-explanatory. A 20-in. manhole was already provided on the top side of the line. The small hole near the bulkhead was cut, through which tools were passed, and to serve as a means of ventilation. The bulkhead was made in four pieces, so mortised that the pieces would slide one upon the other.

The bolts *B* were to be tightened up after the pieces were shoved hard against the wall of the pipe by the jacking screws *C, D, E* and *F*. The holes through which the bolts *B* passed were made large to allow the proper movement of the bulkhead, and were countersunk on one side to allow two thicknesses of 1-in. boards to be nailed, sheet rubber being placed between them.

A groove was cut around the circumference of the head in which packing, rubber or oakum was placed before the head was tightened. The holes were drilled deep enough so that the studs *C, D, E* and *F* could be dropped in while assembling the pieces, when they are screwed into the common center piece, the nuts backed down against the washers and the head tightened.

Nominally, this bulkhead had only a 3-ft. head of water on it, but during a heavy rain it would be as high as 10 ft., owing to back water from a city sewer. It was in place for about three weeks and did not leak. At the same time it was found that another connection would have to be made to a 48-in. supply main, which had a 12-ft. head of water in it.

As in the discharge line, the connection was to consist of a 30-in. saddle flange and 30 ft. of pipe and fittings, and the same conditions were imposed; that is, the line could only be worked on about three hours each night, and it would have to be drained each time, this meaning the draining of 1100 ft. of 48-in. pipe.

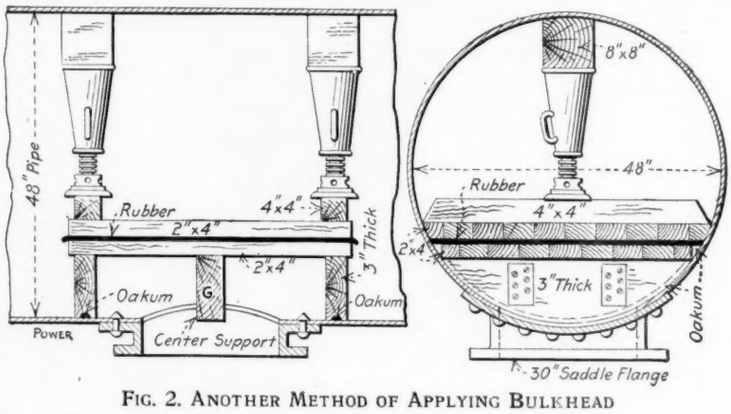


FIG. 2. ANOTHER METHOD OF APPLYING BULKHEAD

Another condition was that there were three engines served from the end of this pipe, and as the flange was to be placed ahead of the condenser connections, some method had to be devised to work on the line and yet not shut the water off of the end of the line.

In Fig. 2 is shown how the job was done. This sketch is self-explanatory; the only thing to look after is to see that the bridge *G* consists of a good piece of wood and arched out over the proposed hole, to allow for putting in the bolts or rivets. The pieces were put in through a 20-in. manhole and assembled inside.

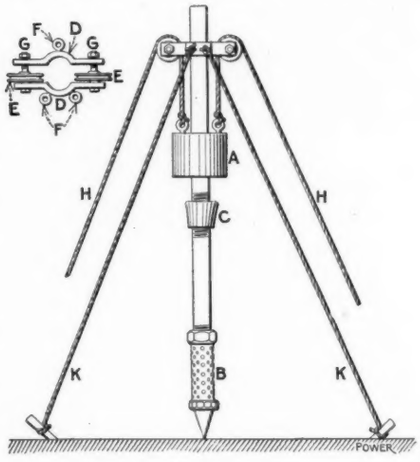
EDWARD H. LANE.

Kansas City, Mo.

Rigging for Driving Small Pipe Well

The engineer is sometimes called upon to drive small pipe wells, and the usual way is to rig up a platform of some sort upon which one or two men can stand and drive the pipe down by means of heavy mauls, but this method is slow and expensive in comparison with the plan that I once saw adopted.

The pipe was cut in short lengths, so as not to have to raise the weight *A*, shown in the illustration, too far from the ground at the start. The drive point *B* was screwed on to the end of a piece of pipe which was coupled to a second piece by means of the special coupling *C*. The weight was then placed in position against the coupling. Then the clamp *D*, which carried the sheaves *EE* and eye-bolt *F*, was placed on the upper end of the pipe, as shown, and held in place by the bolts *GG*.



PIPE-WELL DRILL

The two ropes *HH* were then passed around the sheave wheels and secured to the eye-bolts in the top of the weight, as shown. The whole was then raised to a vertical position by means of the guy lines *K*, which were secured to stakes driven in the ground.

The weight *A* was raised and let drop by two men pulling and letting out on the ropes *HH*. The pipe is kept plumb, by taking up on the guy lines.

A tripod can also be used in place of the guy lines, with a guide at the top in which the pipe can slide. The weight should weigh from 200 to 300 lb., according to the size of the pipe.

GEORGE J. LITTLE.

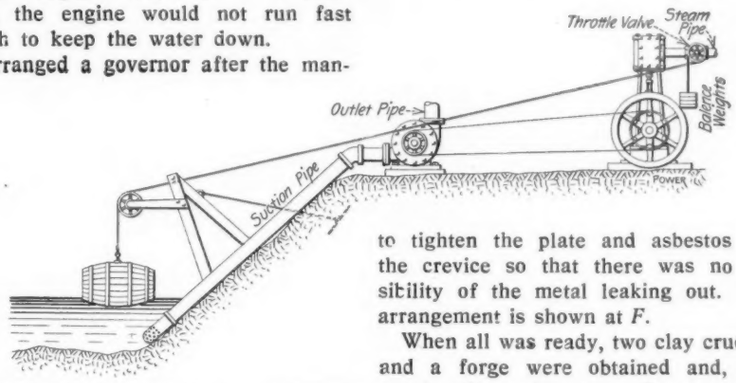
Passaic, N. J.

Barrel Engine Governor

I took charge of a pumping plant at a cement works, and in order to get the marl for making the cement it was necessary to pump the water continually from a large lake so that the men could shovel the marl from the bottom. Owing to the many springs it was necessary to pump 24 hours a day and seven days a week.

I had a belt-driven rotary pump driven by a 20-hp. slide-valve engine. Because of the variation in the amount of water supplied by the springs, and to the engine having no governors, I had trouble in controlling the speed. Sometimes it would drain all the water out of the hole. The pump would then lose its priming and the engine would race. At other times, the engine would not run fast enough to keep the water down.

I arranged a governor after the manner shown in the illustration. It is made out of an ordinary oil barrel with the bung hole plugged. I then secured a light cable to the barrel and ran it over a grooved pulley which was supported by a frame. The other end of the cable ran to the throttle of the engine, on which I fixed another grooved pulley. The cable was wound around this wheel a couple of turns and a weight to counterbalance the barrel was attached to the loose end of the cable.



BARREL FLOAT GOVERNED THE ENGINE SPEED

The device worked finely and was the means of ending my troubles.

J. R. HARDISTY.

Paris, Ont., Can.

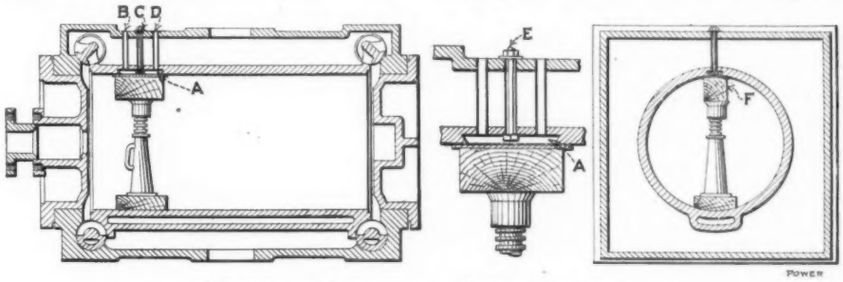
Patching a Corliss Cylinder

Some time ago I had occasion to overhaul a large Corliss engine, and found a deep grove about 2 in. wide and 12 in. long in the top of the cylinder and about 1 ft. from the crank head, as shown at *A* in the illustration.

I finally concluded that a copper patch would be the most durable and best from all standpoints. The groove was squared and chipped and the edges dovetailed, making a slot 2½ in. wide, 12¾ in. long and ⅞ in. deep. I then took a ratchet and drilled three holes, *B*, *C* and *D*, through the top of the cylinder wall and the top of the steam chest. In each of these holes I tightly fitted a piece of ½-in. pipe just long enough to come even with the top of the steam chest and in the cylinder wall even with the bottom of the groove.

A bolt with a square, flat head *E* was made and suspended by the nut and a washer through the pipe in the hole *C* and served as an additional anchor to the dovetail to hold the patch in place. A piece of heavy asbestos paper large enough to more than cover the patch all around was obtained, also a piece of sheet steel of the same size. These were bent to the circumference of the cylinder. Next, a block the same size as the piece of steel was trimmed so as to conform to the same circle. A jack screw, with a block underneath, was used to tighten the plate and asbestos over the crevice so that there was no possibility of the metal leaking out. This arrangement is shown at *F*.

When all was ready, two clay crucibles and a forge were obtained and, after constructing a temporary furnace out of firebrick around the forge, the copper was melted; and, using the hole *B* to pour through and hole *D* as a vent, a good job was obtained. A heavy air hammer was then used to peen the surface of the patch until it was nearly flush with the surrounding surface of the cylinder. It was then scraped smooth.



METHOD OF REPAIRING A SCORED CYLINDER

The melted copper filled each of the pipes *B*, *C* and *D*. It was peened over in the case of the pipes in the holes *B* and *D* and the nut and washer on the bolt took care of the hole *C*.

Many engineers asked me why we did not rebore and bush the cylinder, but it is evident that this would be impracticable, as it would have required a bushing ⅞ or 1 in. thick, together with the necessary time and expense to do the job.

Several mechanics argued that the copper patch should have been peened in in sheets or laminations in order to get a good, tight job, but I concluded that the melted copper would come nearer finding all of the uneven places, and it did.

THOMAS M. STERLING.

Middlebranch, Ohio.

A Troublesome Feed Pump

Some time ago I was called in to examine a feed pump of a neighboring plant which the engineer in charge had been unable to operate satisfactorily. The pump was connected up in the ordinary manner with a globe and check valve next to the boiler and the pump would be operating nicely but at times would not force water into the boiler.

I shut the pump down and soon found that the gasket under the valve plate did not extend across the center rib and both ends of the pump were connected through the opening left under the valve-seat plate. This was remedied and I supposed that I had overcome the trouble, but I found that I had patted myself on the back too soon.

When the pump was started up it ran nicely for a time and then failed to deliver water to the boiler. The cylinder head on the water end was then removed and the plunger repacked. When the pump was again started it pumped water, but later on would not supply a sufficient amount to the boiler.

This "off-again-on-again" action was puzzling, and as no defect in the pump could be found a machinist was called in, but without results. Finally an injector was piped up, in order that the plant could operate while the pump was being given a thorough overhauling, and to my surprise the injector failed to operate.

This was clue enough, however, to the probable cause of the trouble. The cap of the check valve was removed, but it was in good condition; then the globe-valve bonnet was taken off, and it was found that the disk had come off the stem and rested upon the seat, effectually preventing the free admission of water to the boiler.

Just why the disk shifted enough to allow the pump to feed the boiler at one time and then changed again and cut off the water is a matter that has puzzled all concerned.

A. J. BERLIN.

Veedersburg, Ind.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Laying a Plant Up

Replying to Mr. Goal's article, "Starting a Plant Shut Down for a Year," in the issue of Feb. 13, I would suggest when a plant is to be shut down for six months or a year, that all the manhole and handhole plates be taken off the boilers, and all the joints on the boilers and plates be cleaned and covered with heavy oil or hard grease and the boilers thoroughly cleaned.

An outside inspection of the brickwork should also be made, and if any brickwork or resetting be necessary, it should be done at once, to allow time for setting and drying thoroughly, as brickwork is apt to crack if not thoroughly dry when fire is started in the furnace. All valves should be opened wide and then turned back one turn to keep them from sticking. Be sure there are no pockets for water to stand in, freeze and cause trouble.

The water glasses should be taken out so that new rubber gaskets will be put in when starting up as the old rubber will get hard and crack.

The feed pumps should have all the packing taken out of the piston and valve rods, and the stuffing-boxes should be filled with clean waste soaked in cylinder oil; take out the valves, clean and give them a coat of hard grease or cylinder oil, and be sure to leave no water standing in the pumps. The feed-water heater should also be cleaned out.

The engines should be laid up before the boilers; have a light fire in the furnace and very little steam pressure, then the packing can be blown out from around the pistons and valve rods. After this cut the steam off at the boilers and open all valves on the steam lines and engines. This will leave the engine and pipes hot to dry out. When the engine is cold, take off the cylinder head and steam-chest cover; also take out the valves and give the interior of the cylinder, the steam chest and the valves a coat of heavy cylinder oil. This will keep them from rusting. Fill the stuffing-boxes and the pumps with waste and cylinder oil. All bright work should be slushed with grease or heavy oil.

If the plant is in a building which is to be closed up, there should be some means for the air to circulate in the building; if not, the machinery will sweat in the change of temperature from summer to winter. If the smokestack is of iron, it should be covered.

GEORGE L. BAILEY.

New York City.

Taking Charge of a Plant

The article starting up a plant in the Feb. 13 issue, by Mr. Goal deserves some comment. An engineer should take nearly all of the precautions mentioned on first taking charge of a plant, even though it has not been shut down. He should enter the boiler at the earliest opportunity and learn its condition, examine the brickwork and repair it if necessary, replace the defective grate bars, and be sure the steam gage and safety valve are in proper condition.

Standing a year may have loosened some scale and permitted some rust to accumulate, but it would not change the position of the feed pipe or the water column, nor fill the heater with scale or change the setting of the engine valve. A plant closed down yesterday may be in as bad a condition as one of a year ago.

If an engineer knows that the man who formerly had charge was careful and intelligent, he may take a little more time before giving the plant a thorough inspection, but he must know for himself the condition of the steam gage and water column, that the pipes leading to the water column are not clogged, and he will learn about the grate bars the first time he throws coal into the furnace. In short, an engineer should never take things for granted.

G. E. MILES.

La Jara, Colo.

Home Made Damper Regulator

Martin McGerry's damper regulator, as shown in the issue of Feb. 13, is out of true or incomplete.

It can readily be seen from his sketch how the damper can close; in fact, by all the known laws of nature it should have been shown closed; but how it is to open again is a mystery which Mr. McGerry neglected to explain.

If the long pipe, shown as coming from the boiler, entered the top of the cylinder, and the drip pipe entered the bottom, and a steam-tight piston was used instead of the leather cup on the plunger-rod, the device might be made to open and close the damper in a more or less satisfactory manner.

As a rule, I think that home-made damper regulators are unsatisfactory, and that the closeness of regulation and the dependence which can be placed on the operation of the many familiar makes of regulators on the market will amply repay the comparatively small cost of securing one of them.

JOSEPH KING.

Lynn, Mass.

Changes Necessary

The defects of Mr. Gillmore's diagrams shown on page 235 in the Feb. 13 issue can be remedied in the following way:

The governor should be advanced to give earlier admission, as it cannot be done by changing the rods without interfering with the speed regulation. In order to find the amount of advance, turn the engine to the crank-end center, take up all the lost motion in the eccentric straps and rod heads, remove the bonnet from the crank-end steam valve, that the port marks on the valve and the valve seat can be seen, and note the position of the valve. It should have ¼ in. opening or lead.

There is a chisel mark on the hub of the eccentric which should be in line with one on the boss of the governor wheel. If it is not, the two setscrews, acting as stops for the governor crossheads, have to be adjusted. Next loosen the governor springs and throw the weights out, making sure that the eccentric moves to the mark, which denotes its position of least travel.

The valve should lap the port now $\frac{1}{32}$ in. The motion of the valve, due to the change of the eccentric from greatest to least stroke, should therefore be $\frac{9}{32}$ in. for engines up to 18x24 in. The larger sizes require $\frac{9}{32}$-in. lead and the valve motion must be $\frac{5}{16}$ in. If this motion is less, as it seems to be in Mr. Gillmore's engine, the governor must be turned in the direction of rotation until the required lead and lap can be obtained.

An offset key will be necessary. The head-end admission valve must also be looked over while the crank is on the head-end center. The valve motion will be a little greater on this end, owing to the angularity of the eccentric rod, and the position of the eccentric pivot pin.

The right-hand diagram shows that steam leaks into the cylinder after cutoff.

The valve and valve seat should be examined and fitted to a bearing. The compression is late for this style of engine, especially in the left-hand diagram, where the release is also late.

The exhaust valves should close for compression when the crosshead is one-sixth of its travel from the end of the exhaust stroke and open for release when it is one-twelfth from the end of the steam stroke.

M. WERNER.

Chicago, Ill.

I would like to submit the following to J. D. Gilmore, who showed some diagrams in the Feb. 13 issue. Both diagrams show late admission, and the left-hand one shows little or no compression.

As the remedy, I would adjust the steam rods so as to give the valves more lead until the admission line is vertical; adjust the exhaust rods to close the exhaust valve earlier in the case of the left-hand diagram.

J. P. COLTON.

Ohio City, Ohio.

Water Gage Illuminator

Mr. Lehman's letter under the above heading in the Feb. 6 number is interesting. It shows that he has put some thought into the matter. But, will Mr. Lehman tell us what he is going to do with that mirror after the glass lets go?

A little cleanliness is the only condition I ever thought necessary for the easy reading of any water glass that I ever saw.

WILLIAM E. DIXON.

Malden, Mass.

Christie Air Steam Engine

In the Jan. 30 issue, Fred Boone makes several mistakes. He says that if air diffuses from the engine cylinder into the boiler, the diffusion through the inlet valve of a gas engine would fill the engine room with dangerous vapors. He neglects the difference in pressure of the gases in the two cases. The steam-engine cylinder is open to the boiler when the air is under from 75 to 150 lb. pressure per square inch, but the gases in the gas-engine cylinder are at or below atmospheric pressure when the inlet is open.

Further, the air diffusing into the steam pipe must be collected there or returned to the engine while any gases escaping from the inlet of a gas engine are diffused through the atmosphere and are not noticed. The magnitude of the work of diffusion in the atmosphere is well shown by the fact that the atmosphere at Pittsburg contains only one or two parts in 10,000 more CO_2 than the atmosphere in the Rocky Mountains.

Further on Mr. Boone says that the only advantage of mixing air and steam is the lowering of the saturation point of the steam and thus preventing condensation in the cylinder. It does not lower the saturation point of steam to mix air with it. It is true that if the steam and the air pressure are added together, there will be a higher total pressure than that indicated by the temperature of the steam, but the steam will begin to condense as soon as its pressure rises above the pressure of saturated steam for that temperature, regardless of whether there is air present or not.

If air were pumped into the boiler, it would prevent condensation in the steam pipes, but it would do so only because the steam would be generated at a higher temperature and pressure than that at which it is used and, since the pressure would drop without any work being done, the steam would become superheated. The same effect exactly could be obtained by throttling the steam pressure down as it leaves the boiler, but this would be an expensive way of superheating. A common example of this effect is the action of the throttling calorimeter.

The promoters of the Christie engine do not appear to prove that the air-steam is exhausted into the atmosphere at a pressure below the atmospheric pressure. They do prove that the steam is exhausted into the atmosphere at a pressure below atmospheric pressure and this is true in a technical and in every other sense.

GEORGE L. SULLIVAN.

Bozeman, Mont.

One-Man Shifts

An editorial in a recent issue cites the pathetic case of the engineer of the Pasadena electric-light plant who was slowly ground to death in the flywheel pit of an engine, just because the municipality would not stand the extra expense of a second man for the night shift.

A somewhat similar accident happened in a small municipal electric-light plant near Minster, some four years ago, only the engineer was not killed. To the best of my recollection, the plant consisted of five horizontal gas engines, varying in capacity from 35 to 75 hp., all belted to direct-current generators and running on natural gas. The assistant engineer came on duty at midnight, which was the custom, and at about 1 o'clock a small motor which was on a shelf on the wall about 10 ft. above the floor began to spark. The engineer mounted a step-ladder to adjust the brushes and as he did so the stepladder slipped to the floor.

The engineer fell over against a flywheel of one of the engines, breaking an arm and a leg, cracking several ribs and receiving internal injuries which rendered him helpless. He was found in this condition by a tramp who came into the plant at 3 o'clock to seek shelter from the cold. The tramp immediately summoned help and the engineer was taken home, where he lay for six weeks before he was able to get about again.

As the gas engines in this plant were nearly automatic in operation, the plant was kept running without a hitch as the load was light and regular at that time of the night. If it had not been for the timely entrance of the hobo the engineer would have died.

Just how far the practice of one-man shifts in power plants should be carried, is hard to decide. Even if the engineer should decide it, there are still the managers and owners to convince of the necessity of a second man for the safety of the engineer and the operation of the plant.

M. W. UTZ.

Minster, Ohio.

What Caused the Rod to Wear

As to Mr. Kernick's question, in the issue of Feb. 13, the answer is self-evident. The part of the rod remaining in the cylinder is exposed to a mixture of wet steam and air; it may be due to the natural leakage from the throttle, or perhaps from the exhaust pipe. This causes the rod to rust slightly, and when the engine makes a stroke on starting up, this rust is wiped away by the packing in the stuffing-box.

When the engine is shut down the next day a fresh surface is exposed to the oxidation process and, in the course of time, it will become uniformly smaller than that portion of rod not exposed. This might be prevented by swabbing the rod with oil before shutting down.

EDWARD H. LANE.

Kansas City, Mo.

Clearing a Fouled Feed Line

Frank H. Cordner's letter in the Jan. 23 issue, under the above heading, recalls the trouble I once encountered.

The boilers were fed by a pump equipped with extra-hard rubber discharge valves. When the air compressors were shut down, from 6 to 7 a.m. and 12 to 1 p.m., thus depriving the feed-water heater of exhaust steam and allowing the water to cool the valves, would become hard and break. The pieces would be pumped into the feed lines and clog them up.

As a remedy, a thin circular plate was drilled full of small holes and inserted in the flanged joint of the feed line next to the pump. A more convenient arrangement would be a flanged tee, with the branch opening capped, and the strainer located in the flanged joint of

the tee at the discharge end. With such an arrangement the strainer can be easily cleaned after removing the cap.

WILFRED E. BERTRAND.

Philadelphia, Penn.

Home Made Water Heater

In the Feb. 13 issue, John Thorn states that I must have considerable trouble with my home-made water heater, as described in POWER of Dec. 26.

The heater works very satisfactorily at all times, and has been in use for some months past without giving any trouble in any way. He says that he is sure that nothing but steam will issue from the hot-water outlets; he is mistaken as we have hot water of a temperature of about 210 deg. F. at all times. The rumbling noise has never happened since the first time the heater was put into commission. The steam on the line is never below 100 lb. pressure and it is reduced to 45 lb. at the tanks. If the steam pressure was allowed to vibrate below the water pressure then trouble would no doubt happen.

H. L. RUSSELL.

Keams Cañon, Ariz.

Wooden Boxes Cut Shafting

The discussion on sand for hot boxes recalls an experience I had with some bearings in a steam brick dryer which may be helpful to others.

The air circulation was created by a series of butterfly fans driven by a 2-in. shaft running 200 r.p.m. in boxes made of white oak, with the end grain to the shaft.

Some time after the dryer had been in operation I had occasion to examine these boxes and found that the oil holes were filled with something which resembled tarred rope. It was soon ascertained that the wood had charred and was acting as an infinite number of minute cutting tools. The shafting at each box was cut almost in two, the cuttings resembling fine strands of tarred rope.

I renewed the shafting and put in iron boxes, which stopped the cutting trouble.

JOHN T. CHAMBERS.

Kennett Square, Penn.

Does Cold Air Cause Leaky Pipe Joints?

I do not quite agree with Mr. Normand in his letter in the Feb. 13 issue, although his statement that the joint may be leaky and the hot, dry atmosphere absorbs the small amount of escaping steam sounds all right. But what becomes of the hissing noise caused by the escaping steam; does the hot, dry atmosphere eliminate it also?

I have noticed in a number of plants that during the winter months, if the boiler-room doors are left open, the joints will begin leaking and hissing to some extent. After closing the doors the boiler room warms up and the joints stop leaking, though if a practice is made of letting cold air strike the joints, it means making new ones.

The cold air strikes the pipe and causes it to contract, thus shortening its length. The pull comes on the joints, and this to a great extent prevents the bolts from contracting. Contraction and expansion, often repeated, soon start a leak which would not happen if the temperature in the boiler room remained nearly constant.

L. J. OLLIVER.

New Bedford, Mass.

An Inexcusable Loss

Under the above heading, in the Feb. 13 issue, Roscoe S. Helvie gives some figures showing the loss incurred by unnecessarily increasing the clearance of steam engines by using elbows, long nipples, etc., between the cylinder and the cylinder cocks.

After the engine has taken up its load it would seem to me that the cylinder cocks, or drain valves, could be closed, and that the nipples and elbows would in a few seconds become filled with the water of condensation and stay filled, until the engine was stopped, and the cylinder cocks again opened. In this case the clearance would not be much affected.

B. COPPING.

Oliver Springs, Tenn.

Joining Leather Belting

My experience has been that wire lacing is the most satisfactory. The ideal belt is the endless belt, when there is some way to take up the stretch, but this is seldom the case except with electrical machinery.

But any old kind of wire will not give good results; it must be suitable for lacing. (The kind we use is not advertised in POWER, but it should be.) There are four sizes of wire which may be used to mark off the holes for 2-in. belts or over, start ¼ in. from the side and punch a hole every ½ in. The holes should not be much further back from the ends of the belt than its thickness, or a long joint will result which should be used only in endless belts. The punch should not exceed twice the diameter of the wire used.

A good rule for finding the length of the wire is to multiply the width of the belt by 7 for single belts, and by 9 for double belts. We have belts laced this way which are from 1 to 24 in. wide and run on pulleys with diameters of from 3 in. to 24 ft. Our main engine belt is 24 in. wide for a 24-ft. flywheel making 80 r.p.m.

Where new holes have to be made, the lacing may have to be renewed in 9 or 12 months, but the second lacing can be sunk below the face of the belt; then there will be no wear on the wire and it should last until the belt has to be taken up.

Belts over 6 in. wide should be laced in sections. For our 24-in. belt we cut the wire the same length as for four 6-in. belts; this method makes it easier to handle the wire.

S. F. RISE.

Lebanon, Penn.

The British Thermal Unit

Referring to the editorial on "The British Thermal Unit," in the issue of Feb. 27, it is to be regretted that President Bolton, of the American Society of Heating and Ventilating Engineers, did not have at hand the most recent books in which the heat unit is defined. If he had, he would have found that the old definitions in which the temperature of the water is 32 deg. or about 39 deg. are no longer used by good authorities, and that there is a general acceptance of the unit used by Marks and Davis in their "Steam Tables"; viz., the average heat per degree rise of temperature between 32 deg. and 212 deg., or $^1/_{180}$ of the heat required to raise 1 lb. of water from 32 deg. to 212 deg. The reasons for the adoption of this "mean B.t.u." are given in the introduction to Marks and Davis' "Steam Tables and Diagrams," page 6, and they appear to be conclusive.

WILLIAM KENT.

Montclair, N. J.

Indicator Cords Responsible for Diagrams

It is entirely possible, and very likely, that the peculiarities regarding the length of the diagram described by R. A. Cultra in the Feb. 27 issue were caused by the stretch of the indicator cord. This stretch was probably due to the use of guide pulleys (two of which were used) and not to the tension of the additional spring used for convenience in handling the cord.

The head-end diagram taken from an 18x42 Corliss engine was ¼ in. shorter than the other. Separate indicators operated from the same pantograph were used. Presumably, the cord leading to the head-end indicator was between 8 and 10 ft. long.

The frictional resistance caused by guide pulleys greatly increases cord stress, as may be shown by a simple experiment. Fasten a spring balance to a cord running straight from an indicator drum and draw the drum out by hand until it occupies the position assumed at the crank end of the engine stroke. In this position note the force

indicated by the spring balance. Now, pass the cord over the guide pulley attached to the indicator so as to take a half turn around it and repeat the procedure. It will be found that the force necessary to pull out the drum will be between 25 and 50 per cent. greater than in the first case. If the drum is now allowed to return to its head-end position, the force indicated will be less by this same percentage (just before the stroke is reversed) than when the cord runs straight.

Normal values of the extreme forces on the cord under the conditions cited, if no pulleys are used, are 1 lb. at the head and 2 lb. at the crank end. Then the effect of guide pulleys may be to make these ½ lb. and 3 lb. Thus there is lost motion at both ends of the drum stroke corresponding to the stretch induced by 2½ lb. instead of 1 lb.

Regular grades of indicator cord will stretch anywhere between 0.01 and 0.04 in. per foot for each pound. A 10-ft. cord stretching 0.02 in. per pound per foot (a very good quality) would have the effect noted by Mr. Cultra since its total stretch for 1½ lb. would be

$$0.02\ in. \times 10\ ft. \times 1.5 = 0.30\ in.$$

An indicator diagram shortened in this way will be very materially in error at all parts where the lines are not horizontal.

Of course, in the case cited, the shortening may have been due to something else, such as slanting cord or unequal drum diameters, but it is more likely due to the stretch of the cord.

There is a good deal of misconception about the stretch of an indicator cord. It is an elastic material and, like all such, will not stay stretched after being stressed, but will quickly resume its original length.

JULIAN C. SMALLWOOD.

Syracuse, N. Y.

In the Feb. 27 issue, R. A. Cultra asks whether the discrepancy in length of certain diagrams might not be due to the stretch of the indicator cord. No matter how old a cord may be, or how often it may have been used, it will stretch when it is subjected to tension. In proof of this, take a cord, new or old, and stretch it by suspending a pound weight; then after making two ink spots on the cord, 3 or 4 ft. apart, add about 10 lb. more weight.

By carefully tramming the distance between the spots in each case, the degree of stretch may be ascertained. The respective amounts of pull of the drum springs are readily ascertained either by weights or by spring balance, and the tensions in use can be very closely approximated and the relative stretch measured. If the indicators have paper drums and drum pulleys of the same size, then Mr. Cultra may look for the same discrepancy in the length of the diagrams, the cord which stretches the more yielding the shorter diagram. Such measurement of stretch should, of course, include the tension due to the spring which he states was employed for taking up the cord from the pantograph.

Interposition of guide pulleys has the effect of "bunching up" and making the stretch act irregularly, and it is one of the commonest causes for consecutive diagrams showing apparent variation in valve action. This may be the reason for the discrepancy in compression in the diagrams illustrated, by concentrating the effect of stretch in different parts of the respective diagrams. With diagrams taken properly in other respects, it is hardly likely that this discrepancy would exist with any good form of reducing motion, though there is no telling what might happen by employment of a pantograph motion, especially if the indicator cord was fastened, as in the case of the head-end indicator, to one end of a spring which was attached to the cord leading from the pantograph.

After very little use, the joints of a pantograph become worm and loose and are liable to produce crazy and ununiform diagrams, no matter how skillfully the reducing motion may be connected up and used. An additional source of error and uncertainty would arise from making the indicator cord fast to the cord from the pantograph or to the end of that cord, with the latter secured to a spring having one end fast to the floor, as described by Mr. Cultra. No matter how light this spring might be, it would set up some degree of wobbling motion tending to produce a shorter and imperfect diagram.

Though both indicators may take their motion from the same pin on the pantograph reducing motion, Mr. Cultra may expect to obtain shorter diagrams from the indicator connected to the head end of the cylinder, not only because it involves the use of more total length of cord, but also because of the lost motion arising from the wobble of the pantograph cord at the point where the indicator cord and pantograph cord are attached to the spring which is secured to the floor. I would suggest, however, that before attempting any refinement in analysis of the diagrams, he obtain diagrams with a better form of reducing motion than the pantograph. The best forms of reducing motions are none too good.

FRANKLIN VAN WINKLE.

Paterson, N. J.

Shooting Out Keys

I read with interest Mr. Tomlinson's article "Shooting Out Keys," in the Feb. 27 number.

Here in the soft-coal region, where the services of experienced shotfirers are obtainable, a somewhat simpler method is used. A small piece of dynamite is laid against the end of the key to be moved, and in the keyway. It should not, of course, project into the hub, in case the key does not come clear through, unless the hub is to be split. Sometimes clay is laid on the dynamite.

Inexperienced men should begin operations with an exceedingly small charge, gradually increasing it until the results desired are produced. What a piece as long as one's thumb will do is surprising to the uninitiated. Dynamite is, of course, only used as a last resort when mechanical means fail, as they sometimes do on a "hurry-up" repair job.

F. D. BUFFUM.

Ellsworth, Penn.

Automatic Pump Regulator

The automatic pump regulator described by Mr. Lehman in the Mar. 5 issue is rather unusual and, while it no doubt works, it appears to be a sort of "merry-go-round" affair.

Suppose the tank to be full of water and none is being drawn out. With the ordinary pump regulator the pump does not run as long as the tank remains in this condition. If the tank is controlled by Mr. Lehman's pump regulator, as soon as the overflow has stopped, the water drains out of the pail and the weight on the valve lever over-balances the weight of the pail and opens the valve and starts the pump. Water is pumped into the tank on the roof and as the tank is full it runs out of the overflow into the pail, causing the valve to shut and stopping the pump. As soon as the overflow is drained out the pump will start up again and the above cycle will be repeated.

As long as there is a supply of steam to the pump this operation will keep on indefinitely and in a short time enough steam would be wasted to pay for the cost of any one of the many types of automatic pump regulators on the market.

W. L. DURAND.

Brooklyn, N. Y.

Knocking in Air Compressor

In the issue of Mar. 5, page 341, C. R. McGahey says: "Mr. Richards is mistaken when he says that a machine can be run with much variation in the length of the side rods."

I have not made any such statement. In the "straight-line" compressor under discussion, with the crosshead shown in the former sketch, a swiveled crosshead with a long reach between the side rods, slight differences in the lengths of the rods, including the $\frac{1}{16}$ in. mentioned, would make no difference in the running. The crosshead sketched in the issue of Mar. 5 is a different and stiffer affair than that previously shown and is

not an appropriate illustration of the point under discussion.

It seems to me unnecessary to discuss the knockings which result in compressors or engines from slack or lost motion in rods or bearings.

FRANK RICHARDS.

New York City.

Supplementary Exhaust Valves

In the issue of Feb. 20, Mr. Bonn tried to explain the extra exhaust valves I put on an 18x28-in. engine. He would have a very poorly running engine (if it would run at all) if he put them on as his drawing shows.

The pistons are far too short to cover the exhaust ports while the steam acts on the main piston; furthermore, he has the valves too high for economical operation. He states that the engine has been 12 years in service; the fact of the matter is it was built 35 years ago.

The engine runs at 110 r.p.m. and should be able to deliver 150 hp. with 90-lb. gage steam pressure. Whenever the load increased to 100 hp. the speed dropped to 106 r.p.m. To find the engine trouble I bought an indicator outfit for the money I received for selling an old boiler which my employer gave me. I could not detect any leaks after testing the piston and valves. The exhaust pipe was clear. I then measured the exhaust ports; they were 12 in. long and 1¼ in. wide, and they seemed rather small.

More power was wanted in the factory, so I sent some diagrams and the area of the steam ports to the engine company. It would not do anything to the engine; it blamed the piston and valves for the back pressure. I suggested some extra exhaust valves, but the company would not take the responsibility.

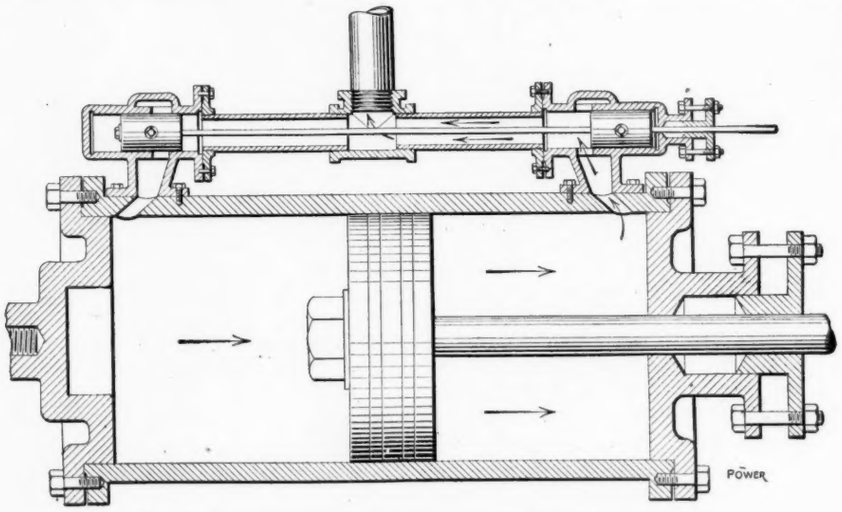


FIG. 1. DESIGN AND ARRANGEMENT OF AUXILIARY VALVES

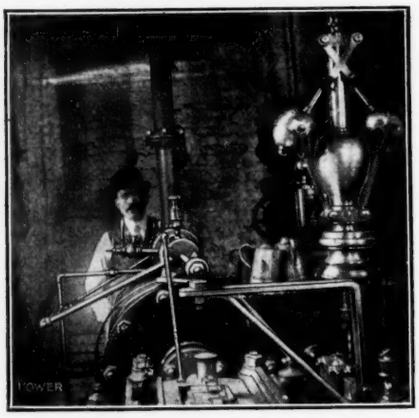


FIG. 2. MANNER OF CONNECTING EXTRA EXHAUST VALVES TO VALVE GEAR

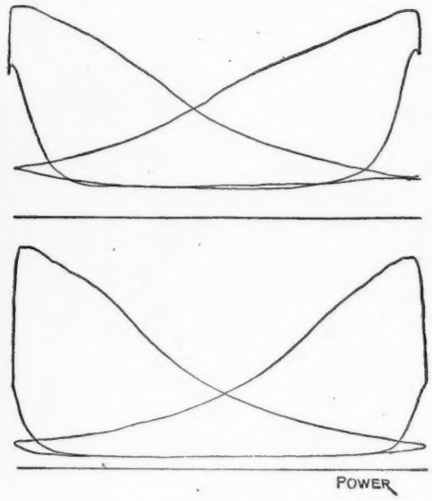


FIG. 3. DIAGRAMS TAKEN BEFORE AND AFTER VALVES WERE PUT ON

Then I went ahead on my own responsibility. I made a drawing and had some castings made by a local machine shop and put the extra valves on top of the cylinder, as shown in Figs. 1 and 2. The diameter of the valve pistons is 3 in.; the slots in the cylinder are 5x1½ in. These valves should have been larger, but I would not risk the chance of weakening the cylinder.

The diagrams in Fig. 3 indicate the conditions in the cylinder before and after the change was made. The back pressure shown by the lower set of diagrams is 4½ lb. There are over 4000 ft. of pipe on the exhaust line in the winter, during which season these diagrams were taken.

Fig. 2 shows how the valves are controlled from the regular eccentric. The engine is running now at 110 r.p.m., regardless of the load, and I have reduced the grate area 4 sq.ft. The machines throughout the factory are running uniformly and everybody, including myself, is pleased.

WENDELIN SENGE.

South Bound Brook, N. J.

See the Hump

Phil Lighte asks for an explanation of the "humps" in a pair of diagrams reproduced in the issue of Mar. 5.

It is evident that these humps are not true representations of a change in pressure inside the engine cylinder and I believe that they are due to lost motion in the indicator. The only part of the instrument influenced by the change of excess pressure from the bottom to the top of the indicator piston is the spring, which, from having been under compression, comes under tension. The pencil, levers and piston simply register the humps, the lost motion being on either or both ends of the spring. Perhaps it may not be screwed up tight.

Although the humps cannot be shown up plainly in the compression lines, on account of these lines being almost vertical where they pass the atmospheric line, the humps manifest themselves there to the same amount.

The result of this lost motion is that the part of the diagram above the atmospheric line is raised the amount which the continuation of one part of the expansion line is above the remaining part. The mean effective pressure shown will be too high.

That four of the diagrams taken showed the humps and the other not, points to the tightening up of the screw and spring without the knowledge of the operator. This may possibly have resulted from the swinging of the pencil arm back and forth toward the drum, while taking the diagrams, which sometimes drags along the cap, to which the top part of the spring is fastened. This will increase or reduce the lost motion in a loose spring, according to the direction in which the cap turns.

RULOF KLEIN.

New York City.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Watts Required for a Car to Ascend a Grade

If a trolley car weighs 8 tons, the passengers 2 tons, and the efficiency of the motors is 75 per cent., how many watts are supplied to the motors when the car is propelled up a 7 per cent. grade at a speed of 12 miles an hour?

G. O.

The total weight of the car and its load is

$$8 + 2 = 10 \text{ } tons$$

and the distance traveled per minute is

$$\frac{12 \times 5280}{60} = 1056 \text{ } ft.$$

A 7 per cent. grade means a vertical rise of 7 ft. in every hundred traveled; therefore the vertical rise in feet per minute would be

$$1056 \times 0.07 = 73.92$$

The foot-pounds of work done per minute by the motors would be

$$20{,}000 \times 73.92 = 1{,}478{,}400$$

and as 1 hp. equals 33,000 ft.-lb. per minute, the horsepower developed by the motors would be

$$\frac{1{,}478{,}400}{33{,}000} = 44.8$$

The watts output of the motors would be

$$44.8 \times 746 = 33{,}420.8$$

As the efficiency of the motors is 75 per cent. their intake would be

$$\frac{33{,}420.8}{0.75} = 44{,}561 \text{ } watts$$

Assuming the tractive force to be 25 lb. per ton, the foot-pounds of work done per minute by the motors if running on the level road would be

$$1056 \times 250 = 264{,}000$$

and the watts delivered

$$\frac{264{,}000 \times 746}{33{,}000 \times 0.75} = 7957$$

The total power required for the car to ascend the grade would be, therefore,

$$44{,}561 + 7957 = 52{,}518 \text{ } watts$$

or 52.52 kw.

Specific Gravity

How is the specific gravity of a solid determined?

C. L. A.

The specific gravity of a substance is its weight compared with an equal bulk of pure water. A solid immersed in a fluid will displace exactly its own volume, and the loss of weight of a body equals the weight of the fluid which it displaces. For example, a solid weighed in air was 60 oz. when suspended from the scale beam and when weighed in a vessel of water it was only 50 oz. Therefore, the weight of an equal bulk of water was

$$60 - 50 = 10 \text{ } oz.$$

and the solid would be $\frac{60}{10} = 6$ times heavier than water.

Changing Direction of Rotation of Engine with Shaft Governor

How can the direction of rotation of an engine with a shaft governor be changed?

W. C. C.

Change the position of the eccentric and all the governor parts. The pivoted ends of the levers should always lead and the weights follow. Engine manufacturers usually have extra bosses cast on the proper flywheel arm, and sometimes drilled to facilitate such a change when desired.

Steam per Horsepower per Hour

What is the rule to find the steam consumption per horsepower per hour?

B. E. D.

The only way to find out how much steam an engine is using per horsepower per hour is to measure both the power and the steam. The power can be measured with a prony brake or an indicator, and the steam by measuring or weighing the feed water if the boiler supplies nothing but the engine. The amount of steam can also be found by condensing and weighing it as it leaves the engine.

Starting Up a Condensing Engine

When starting up a condensing engine with a jet condenser, should the injection valve be opened before or after starting the engine?

M. V. L.

The injection valve should be opened after the engine has been warmed up, and the engine should be started immediately after the valve has been opened. When shutting down, close the throttle first, and when the speed of the engine has been materially reduced, close the injection valve.

Thickness of Cylinder Walls

What is the rule for the thickness of the cylinder walls of an engine?

F. W. K.

Multiply the initial pressure in the cylinder by the cylinder diameter, divide the result by 4000, and add 0.06 to the quotient. Example: Initial pressure = 95 lb. Cylinder diameter = 20 in.

$$\frac{95 \times 20}{4000} + 0.6 = 1.075 \text{ } in.$$

for the wall thickness.

Three-phase Motor on a Two-phase Circuit

Will a three-phase motor run on a two-phase circuit?

D. E. W.

Yes; but not satisfactorily, because the rotating magnetic field would be unsymmetrical and one of the phases would be nonoperative. The motor would have less than two-thirds of the capacity that it would have on a three-phase circuit. If delta-wound, one of the legs of the winding would have to be disconnected from the others.

Liquid Air

Can liquid air be frozen?

J. S.

No. Prof. Dewar found when the temperature of liquid air was reduced to —346 deg. F. it became apparently solid or frozen. But when the solid mass was placed in a strong magnetic field, the oxygen was drawn out toward the poles of the magnet in a liquid form, showing that nitrogen may be frozen at —346 deg. F. The temperature at which liquid oxygen freezes has not yet been determined.

Compressing Oxygen

Can oxygen be compressed in an ordinary air compressor to a pressure of 1500 lb.?

P. G.

An ordinary air compressor will not do because the oxygen attacks the cast iron, and noncorrosive metal must be used. Also oxygen may form a dangerous explosive mixture with lubricants. Special machines are built which will successfully compress oxygen to 4000 lb. per square inch.

Uniform Boiler Specifications

Recommendations by the committee on uniform boiler specifications of the American Boiler Manufacturers' Association, for changes in the Massachusetts boiler laws.

Since its first convention in 1889, the American Boiler Manufacturers' Association has striven for uniform boiler specifications. The first move was directed chiefly toward the material used, and it was largely through the efforts of this association that steel was substituted for iron in boiler construction at that time.

Later the association directed its attention toward construction and workmanship. A committee met at Atlantic City in the summer of 1898 and formulated rules covering in a practical and general manner all the important details of material and workmanship in boiler construction. These rules were officially adopted at the annual convention of the association at St. Louis in the fall of that year.

Subsequently, the association tried to have the use of these rules made imperative in a number of states, but in every case legislative action proved futile, and it was then decided that more could be gained by instituting an educational campaign, with the hope that legislative action would follow.

Two of the states, Massachusetts and Ohio, have already adopted laws along the lines laid down, and at the 1909 session of the American Boiler Manufacturers' Association a general indorsement of the Massachusetts rules was passed, subject to such improvements and amendments as the association deemed necessary.

At the convention in Boston last year a request was made of those members living in Massachusetts to make suggestions for such modifications as their experience would seem to warrant. A committee was appointed and after careful consideration it made a number of recommendations for changes in the present Massachusetts boiler rules.

It was recommended that part 2, section 1, be changed so as to make the allowable pressure, 15 lb. instead of 25 lb. per square inch, on a boiler constructed wholly of cast iron, such boilers to be used only for heating purposes; that heating boilers built entirely of steel have a pressure not exceeding 25 lb. per square inch.

Paragraph 7, section 1, of the Massachusetts rules states that the lowest factors of safety used for boilers the shells or drums of which are exposed to the products of combustion and the longitudinal joints of which are of lap-riveted construction, shall be, for boilers not over 10 years old, 5; over 10 and not over 15 years old, 5.5; over 15 and not over 20 years old, 5.75; 6 for boilers over 20 years old; and 5 for boilers the longitudinal joints of which are of lap-riveted construction and the shells or drums of which are not exposed to the products of combustion.

The committee objects to this and recommends that it be changed, holding that the factor of safety should be based upon the actual physical condition of the boiler at the time of inspection rather than upon its age.

It also disagrees with the general prohibition of cast-iron seats for safety valves, as stated in section 8, holding that the universal satisfactory experience on the navigable waters of the Mississippi valley, with lever type safety valves having cast-iron seats precludes this discrimination.

Paragraph 9 now states that safety valves hereafter installed on boilers shall not exceed 5 in. in diameter and shall be of the direct spring-loaded type with the seat and bearing surface of the disk inclined at an angle of about 45 deg. to the center line of the spindle; and designed with a substantial lifting device so that the disk can be lifted from its seat with the spindle, not less than one-eighth the diameter of the valve, when the pressure on the boiler is 75 per cent. of that at which the safety valve is set to blow.

The committee disagreed with this on the ground that the experience of reputable manufacturers of standard pop safety valves shows the flat seat to be equally safe and effective.

Regarding paragraph 11, it claims that the maximum diameter of the fusible metal of safety plugs should not exceed ⅜ in., whereas the law now states that the least diameter of fusible metal shall not be less than ½ in., except for working pressures over 175 lb., or when it is necessary to place a fusible plug in a tube, in which case the least diameter of fusible metal shall not be less than ⅜ in.

Section 6, paragraphs 1 to 3, of the present Massachusetts rules, relating to the hydrostatic pressure test, states that when a boiler is tested, the maximum pressure applied shall not exceed 1½ times the maximum allowable working pressure, except that twice the maximum allowable working pressure may be applied on boilers permitted to carry not over 25 lb. pressure per square inch or on pipe boilers. Furthermore, when making annual inspections on boilers constructed wholly of cast iron or on pipe boilers a hydrostatic pressure test of not less than 1½ times and not more than twice the maximum allowable working pressure shall be applied; also, the boiler inspector, after applying the hydrostatic pressure test, shall thoroughly examine every accessible part of the boiler, both internal and external.

The committee recommends that this be changed to conform to the American boiler specifications, section 30, which prescribes that the hydrostatic test be made on completed boilers built strictly according to these specifications, is never to exceed the working pressure by more than one-third of itself, and this excess shall be limited to 100 lb. per square inch; the water used for testing to have a temperature of at least 125 deg. F.

In place of paragraphs 1 to 3, section 1, part 3, of the Massachusetts rules, covering the chemical and physical properties of metal to be used, it is recommended that paragraphs 1 to 3 of the Uniform American Boiler Specifications be substituted; these are as follows:

1. CAST IRON—Should be of soft, gray texture and high degree of ductility. To be used only for hand-hole plates, crabs, yokes, etc., and manheads. It is a dangerous metal to be used in mud drums, legs, necks, headers, manhole rings or any part of a boiler subject to tensile strains; its use is prohibited for such parts.

2. STEEL—Homogeneous steel made by the openhearth or crucible processes, and having the following qualities, is to be used in all boilers:

	Tensile strength, pounds
Flange or boiler steel	55,000 to 65,000

"When it is stipulated that the plates are to be flanged, the physical properties shall be the same as required for firebox steel."

	Tensile strength pounds
Firebox steel	52,000 to 62,000
Extra soft steel	45,000 to 55,000

	Elongation in 8 in., per cent.
Flange or boiler steel	25
Firebox steel	26
Extra soft steel	28

Elastic limit to be not less than one-half of the ultimate tensile strength.

	Chemical requirements	
	Sulphur, per cent.	Phosphorus, per cent.
Flange or boiler steel.	0.03	0.04
Firebox steel	0.03	0.04
Extra soft steel	0.03	0.04

For all plates the elastic limit to be at least one-half the ultimate strength; percentage of manganese and carbon left to the judgment of the steel maker.

Test section to be 8 in. long, planed or milled edges; its cross-sectional area not less than one-half of one square inch, nor width less than the thickness of the plate.

BENDING TEST—Steel up to ½ in. thickness must stand bending double and being hammered down on itself; above that thickness it must bend round a mandrel of diameter of 1½ times the thickness of plate down to 180 deg. All without showing signs of distress.

Bending test piece to be in length not less than 16 times thickness of plate, and rough, shear edges milled or filed off. Such pieces to be cut both lengthwise and crosswise of the plate.

All tests to be made at the steel mill. Three pulling tests and three bending tests to be made from each heat. If one fails the manufacturer may furnish and test a fourth piece, but if two fail the entire heat to be rejected.

Certified copies of tests to be furnished each member of the American Boiler Manufacturers' Association from heats from which his plates are made.

3. RIVETS to be of good charcoal iron, or of a soft, mild steel, having the same physical and chemical properties as the firebox plates, and must test hot and cold by driving down on an anvil with the head in a die; by nicking and bending, by bending back on themselves cold, without developing cracks or flaws.

For paragraph 16 of the same section of the Massachusetts rules it is recommended that the table of allowances for overweight on plates be omitted and the following rule substituted: Plates will be considered up to gage if measuring not over 0.01 in. less than the specified gage The committee holds that the paragraphs 1 to 3 of section 2, part 3, relating to the material used in the shell plates, heads and rivets, should be annulled and that the qualities ascertained by actual tests should be conclusive as defining the material to be employed instead of such indefinite designations as "flange," "firebox," or "extra soft" steel, which are the terms at present specified.

Paragraph 7 of the same section now states that mud drums of water-tube boilers shall be of wrought or cast steel when the working pressure exceeds 160 lb. per square inch. It is recommended that this be amended by prohibiting the use of cast iron for mud drums.

The committee recommends that section 3, paragraph 2, of the present rules providing that the stamp (Massachusetts standard) shall be affixed by the inspector, etc., should not be construed to compel the manufacturer to stop or retard work in process or prior to completion on account of the temporary absence of the inspector.

It is further held that paragraph 6, section 4, which limits boilers of lap-riveted construction to 100 lb. per square inch and to a diameter not exceeding 36 in., is not warranted by the experience of reputable boiler manufacturers. Lap-riveted boilers 48 in. in diameter have been and are in successful use on the Mississippi River.

Also, it is suggested that paragraph 10, section 4, which provides that the thickness of plates in a shell or drum shall be of the same gage, should be modified so that a manufacturer while using plates of sufficient gage under these rules may employ them with other plates of a somewhat thicker gage which he happens to have in stock.

In paragraph 16, which deals with bumped heads, it was found that there has been considerable misunderstanding in regard to which is a convex head and which is a concave head. It is suggested that these heads be designated as "externally convex" and "externally concave." Furthermore, it is suggested that 0.8 be substituted for 0.6 as the constant in the formula for the minimum thickness of an externally concave head.

$$t = \frac{R \times F.S. \times P}{0.6\ (T.S.)}$$

in which R is one-half the radius to which the head is bumped; $F.S.$ is the factor of safety; P the working pressure; $T.S.$ the tensile strength as stamped by the manufacturer, and t is the thickness of the head in inches.

To avoid misunderstanding it is recommended that paragraphs 17 and 18 be combined as one so that it may be thoroughly understood that the convex or concave head, which has a manhole, shall have an increased thickness of ⅛ in. and a turned-in flange of a depth not less than three times the thickness of the head.

As to paragraph 19 it is held that the word "flat" should be omitted so that the minimum thickness of plates in all stayed surfaces shall be $\frac{5}{16}$ in.

The committee would so amend paragraph 21 that an increased pitch of not over 1½ in. be allowed in the staybolt adjacent to the furnace door or handhole opening, but in one direction only.

In paragraph 23, relating to staybolts, it is recommended that for small heating boilers limited to 25 lb. and under, no staybolts be required. Also, in paragraph 25, to avoid misunderstanding, it is suggested that the language be so amended that the pitch of staybolts shall be measured on the inside of the furnace.

For paragraph 32, it is contended that the stays specified on each side of the manhole in horizontal return-tubular boilers should not be required to pass through the front head because of the difficulty in keeping these joints tight; and that other equally strong means of staying may be adopted.

The committee believed that paragraphs 40 and 41 relating to cast-iron nozzles and flanges should be placed in harmony with the present manufacturers' standard.

Paragraph 43 states that rivet holes, except for attaching stays or angle bars to heads, shall be drilled full size with the plates, butt straps and heads bolted in position; or they may be punched not to exceed ¼ in. less than full size for plates over $\frac{5}{16}$ in. in thickness, and ⅛ in. less than full size for plates not exceeding $\frac{5}{16}$ in. in thickness, and then drilled or reamed to full size with the plates and heads bolted in position. The committee believes this should be amended so as not to apply to holes in the fittings.

It is recommended that paragraph 45 be eliminated, because it cannot be complied with in the case of rivets driven by air tools or by hand, which must always be employed where hydraulic riveters cannot reach them; and a rule should not discriminate against smaller boiler shops unable to use hydraulic riveters.

It is suggested that paragraph 48 be amended by omitting everything after the word "chamfered," so that it shall read: "The edges of tube holes shall be chamfered."

In paragraph 49, the word "carefully" is believed to be better than "substantially"; the rule will then read: "A fire-tube boiler shall have the ends of the tubes carefully beaded."

It is held that paragraph 53 should be amended to allow the use of pressed-steel and wrought-iron nozzles in addition to those specified.

The word "oval" is suggested instead of "elliptical" in designating the shape of the manhole opening, paragraph 54.

Paragraph 59, to be amended to permit the use of cast-iron manhole plates which are universally used in the boilers in the Middle West.

An amendment is suggested for paragraphs 60 and 61 so as to make it optional whether a manhole or a handhole shall be used in the front head of horizontal tubular boilers.

The committee recommends that paragraph 62 be eliminated, because it is a very difficult and precarious thing to keep a handhole in good order in the rear head of a horizontal tubular boiler below the tubes.

The requirement of a handhole near the throat sheet of a horizontal tubular boiler should be eliminated by striking out the words "also where possible one near the throat sheet," paragraph 63.

In paragraph 65, that ¾-in. be substituted for the 1-in. pipe size.

Paragraphs 67, 68 and 69, which definitely prescribe the hanging or support of large tubular boilers, should be eliminated and manufacturers permitted to follow their present practice which has been proved safe.

It is recommended that paragraph 70 be struck out, and paragraph 31 of the Uniform American Boiler Specifications be substituted.

Section 5, paragraph 3, should be changed from 2 in. to 3 in.

Paragraph 5 is sometimes misunderstood, and should be rewritten to make it clear that two valves are required to be placed on any individual steam pipe leading from a boiler to a steam main or to another boiler, in the interest of the safety of any man entering the boiler

while the other boiler or boilers are in service.

The committee believes that 100 lb. should be substituted for 135 lb. in paragraph 18.

Regarding paragraph 23, it is suggested that the words "except such portion of the piping as is exposed to the gases of combustion" be added.

The committee believes that paragraph 31, relating to the support of boilers, should be placed under section 1, part 2, referring to cast-iron boilers.

That in paragraph 3, of section 6, the sentences (a) and (b) be cut out, as they discriminate against lap-riveted joints which are in satisfactory use in thousands of boilers in the Middle West. Also that sentences (d) and (e) be eliminated, and that the factor of safety be based on actual conditions of the boilers instead of their age.

It is further recommended that the last sentence covering the hydrostatic-pressure test should be struck out and that paragraph 30 of the Uniform American Boiler Specifications be substituted.

Acme System of Controlled Combustion

The Acme system of controlled combustion is designed to maintain a highly heated zone for the union of the liberated gases with a proper amount of oxygen, to retain the gases of combustion in contact with the heating surface of the boiler the greatest possible time, and to confine the variability of load service to a part of the boiler plant. Details of the system are shown in Fig. 1.

The plant consists of an engine-driven blower outfit, the speed of which is controlled by the variation of steam pressure in the boilers, and the fan delivers air at any desired pressure to the ashpit of the furnace. A means is also adopted for adjusting the relative air supply and outflow of waste gases so as to obtain a determined rate of combustion while maintaining practically atmospheric pressure in the combustion chamber. This arrangement also prevents the influx of cold air at the furnace doors during the firing period, or through cracks in the setting.

The Acme system of controlled combustion accomplishes an economy in the operation of the steam plant by supplying the greater portion of the entire load demand of the plant from a number of the boilers. Each boiler of the constant group is operated under such fixed conditions of combustion as may be best suited to its particular requirement. The remaining or fluctuating portion of the load demand is supplied from a further number of boilers which are variably operated for this purpose. In this manner a major portion of the total vaporization requirement of the plant is supplied from an appropriate number of boiler units, operated under conditions approaching as closely as possible to maximum efficiency. The losses, therefore, attendant upon the regulation of combustion to meet fluctuations in the load are confined to a comparatively smaller number of variably operated units and are correspondingly reduced.

The regulation of the boilers which serve as the constant supply units is obtained by introducing a constant de-

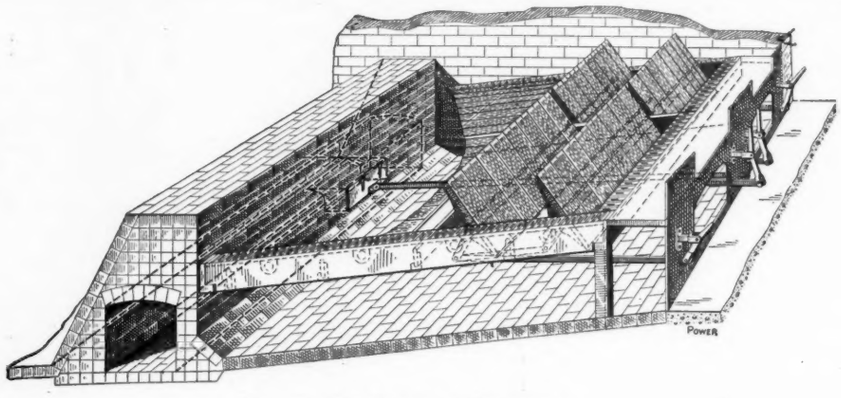


FIG. 2. AIR DUCT IN BRIDGEWALL

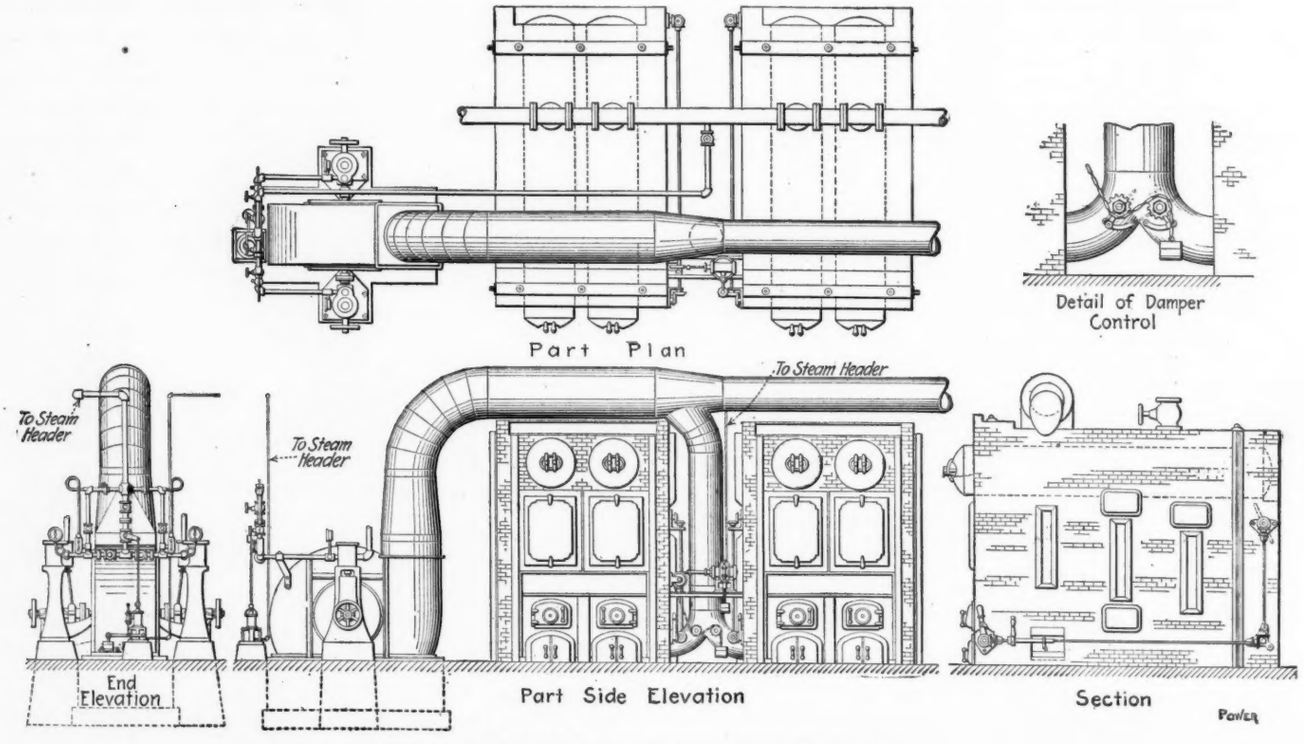


FIG. 1. ACME SYSTEM OF CONTROLLING COMBUSTION

livery of forced air into the ashpit and adjusting the stack dampers so that the approximate atmospheric furnace pressure is normally maintained.

The regulation of the variable supply units is accomplished by varying the amount of forced draft admitted to their ashpits, the stack dampers of such variable units remaining in a fixed position of adjustment which is suited to their normal or average requirement. The air duct to the ashpit is shown in Fig. 2. In the event of a change in load exceeding the limits of range served by the variable units, a general regulation of the constant supply units to meet the emergency is provided, the general regulation being effected by varying the force of the draft by retarding or accelerating the speed of the blower.

The several boilers are supplied with gearing interconnecting their respective stack and ashpit dampers so that by the operation of a single lever, both dampers may be simultaneously operated. The interconnecting gearing is supplied with an adjustable connection between the dampers whereby the relative positions of the two may be shifted as desired, and their simultaneous operation effected accordingly; so that with a normal or predetermined delivery of forced air, the operation of each unit will approach as closely as possible its maximum efficiency, which condition may be determined by actual test. The rate of combustion in such units will be practically constant, and variations in such operation when produced by the manipulation of the damper operating handle will maintain the same furnace pressure.

The range of the control of the variable units is fixed so as to include all such normal fluctuations of the load as occur during usual or ordinary conditions. A range of 5 lb., with an adequate number of variable units, is ordinarily sufficient to accommodate usual running fluctuations, but such range may be adjusted according to the judgment of the operator. The valve controlling the general regulation is set to respond only to changes in steam pressure exceeding the range served by the variable units, and when thus called into operation regulates the supply of forced draft to the ashpit.

The determination of the relative numbers of constant and variable-supply units for any given steam plant will depend upon the ordinary conditions of running. The constant-supply units will preferably be sufficient in number to carry the maximum constant portion of the load, while the variable units will be sufficient to perform the balance within reasonable limits of control. On occasions, when the capacity of the plant is desired to be materially or permanently changed or the range of normal fluctuations is increased or diminished, it will be necessary to alter the relative numbers of constant and variable units.

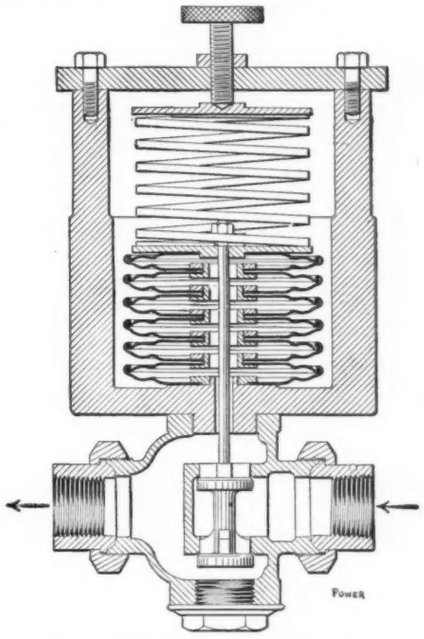
A steam-controlling valve is placed at one end of the battery and operates a shaft running to the rear of the boilers, having an adjustable connection to each of the dampers in the uptake. This valve is set to close at about 5 lb. above the normal steam pressure and is so adjusted that a desired pressure in the combustion chamber may be readily obtained.

A second valve is placed on the other end of the battery with a shaft traversing the first one and means of connection with any of the individual dampers. This valve is set at normal steam pressure and regulates the draft on the control boiler or boilers. A simple attachment is arranged so that each damper may be closed by hand during the process of cleaning, or when putting a boiler out of commission.

This system is manufactured by the Acme Furnace Equipment Co., 39 Cortlandt St., New York City.

Ziermore Pressure Regulator

The improved Ziermore pressure regulator is illustrated herewith. All parts are made of bronze with the exception of the diaphragms which are of silver and german silver. The connecting fittings are made male and female and form a part of the diaphragm. Due to this construction it is possible to build as long a battery of multiple diaphragms as is desired to suit the throw of the valve, and also to add to or take from a diaphragm from the battery.



ZIERMORE PRESSURE REGULATOR

The lower end of the battery of diaphragms is anchored to the threaded lug of the yoke, but the upper end is free to move with the pressure on the inside of the diaphragms. To the upper end of the diaphragms is attached a bronze plate to serve as a spring support, a shoulder for the diaphragm and an anchorage for the valve rod.

The valve rod passes through the battery without friction. The upper end of the rod is anchored to the movable end and the lower end of the rod to the valve disk, thus causing it to move toward or away from its seats as the battery inflates or deflates with any change of pressure within them.

The spring tension is set against the top of the battery of diaphragms by means of the adjusting screw and the reduced pressure is thus produced and controlled.

The fluid passes through the regulator as is indicated by the arrow. The low pressure only has access to the battery of diaphragms and enters them from the low-pressure side of the regulator, through the same port which the valve rod enters.

Any change of the low pressure will cause the battery diaphragms to move, and the valve disk must move exactly the same distance.

The regulator may be placed in service in any position without interfering with its sensitive action. It is suitable for controlling steam, gas, air, water, oil and many kinds of acids. Wire-drawing is almost eliminated, and chattering is entirely unknown, it is claimed. This regulator is manufactured by the Ziermore Regulator Co., Johnsonburg, Penn.

Tubes Blow Out

Two tubes of a boiler in the Fox Co. paper mill at Cincinnati, Ohio, blew out on Mar. 3 and fatally scalded two men. It is alleged by State Inspector Huddleston in his testimony before the coroner that the boilers in the mill were not properly cared for and that proper safety precautions were not provided for the employees' safety. The boiler was said to be carrying 75 lb. pressure, about one-half the regular working pressure. A tube blew out of the same boiler on Feb. 29 and caused some damage.

The Uptodateness of Siam

John Barrett, director of the Bureau of American Republics, was once minister to Bangkok, Siam, where he had a chance to observe the Siamese method of doing business.

Shortly after the electric-light plant was installed, complaints were sometimes received from dissatisfied consumers, and the manager sent out the following on a post card:

Sir—For the case that your electric light should fail, we beg to send you with this a return post card, which please send us at once when you find your light out. The company will then be most happy to send you another post card. Yours truly, Manager Siam Electricity, Limited.

As an evidence of Oriental belief in Occidental methods of public utility, the foregoing letter is a fair sample. But, taken literally, its offer to forward a second card to the subscriber leaves one in the dark as to the quality of the light service provided by the Siam Limited.

Saves Money on Lights

It is estimated by Henry Berg, chief engineer of the Orange, N. J., municipal lighting plant, that the city has saved $10,000 in the first year of the plant's operation. It would have paid the Public Service Electric Co. at the rate of $85 a lamp, whereas it cost $60 a lamp under municipal operation.

The net cost of operation was $17,997.12. The city has 369 arc lamps and 156 tungsten lamps, an equivalent of 408 arc lamps. Thus the cost of operation and maintenance, distributed among that number of lamps would be $44.11. Interest, sinking fund, and depreciation accounts would bring the charge per lamp up to $60.

Oil Mill Superintendents' Convention

On June 12 to 14, the National Oil Mill Superintendents' Association will hold its annual convention at Memphis, Tenn. There will be a large exhibit of trade supplies, and cotton-oil and other mill machinery.

Low railroad and hotel rates will be made for the delegates and friends and intending exhibitors can obtain full information by writing to F. M. Smith, of the Oriental Oil Co., Dallas, Tex.

The Oil Mill Machinery Manufacturers and the Supplymen's associations will meet in Memphis at the same time.

NEW PUBLICATIONS

THERMODYNAMICS OF THE STEAM TURBINE. By C. H. Peabody. Published by John Wiley & Sons, New York. Cloth; 282 pages, 9x5¾ in. Price, $3, net.

The author's preface states that this is an attempt at a systematic treatment of the thermodynamics of the steam turbine for students in technical schools. The author's ability is sufficiently well known to make it easy to comprehend the degree of success with which the "attempt" has met.

Constructive details are considered so far as they are necessary for a proper understanding of the thermodynamic computations or as they are related to such computations. The methods in general are those accepted by steam-turbine designers, but certain methods have been devised by the author either to make them more complete or to provide more rapid and precise determinations of conditions and proportions.

A 13-page introductory chapter deals briefly with definitions, standards and units and is included for the convenience of reference and to avoid ambiguity.

Successive chapters deal with steam nozzles; jets and vanes; simple impulse turbines; pressure compounding; velocity compounding; pressure and velocity compounding; reaction turbines; accessories; effect of conditions, and marine steam turbines.

GAS ENGINES. By W. J. Marshall and H. Riall Sankey. Published by D. Van Nostrand Co., New York. Cloth; 280 pages, 5½x8 in.; 127 illustrations. Price, $2.

The avowed purpose of this work is to inform those who, "being either purchasers or users of gas engines, wish to know the principles underlying their design and construction, the methods of diagnosing defects when they occur, and the steps to be taken to remedy such defects." This purpose is well executed in general, although some errors and arbitrary statements have crept in which are more unfortunate, in view of the chosen audience, than they would be in a college textbook or a reference book. On page 2 it is stated that the density of water is maximum between 62 and 63 deg. F. On page 3 and elsewhere, absolute zero is put at 461 deg. F. and 273 deg. C., which values are inconsistent with each other. On page 4, Boyle's law is incorrectly stated, the word "inversely" having been omitted after the word "varies" in the fourth line. On page 19 the release temperature is put at 1400 deg. F., which is rather excessive. On page 20 an engine showing 30 per cent. indicated heat efficiency is classed as a "bad engine" and a "good engine" is accredited with an efficiency of 40 per cent. We should be glad to see an internal-combustion engine, outside the Diesel class, showing in everyday service even the "bad" efficiency cited. On page 24 the average low heat value of pressure (bituminous) producer gas is put at 155 B.t.u. per cubic foot, that of suction producer gas (anthracite) at 146 and that of suction producer coke gas at 135 B.t.u. On page 81, the primary wipercam levers are drawn upside down, as regards the common practice. On page 95, the deflector is of about twice the normal proportions and located in the center of the piston face, which is never true in practice.

Chapters II to VI of the book are commendable, but the first chapter and the last seven, dealing successively with elementary principles, cooling, ignition, engine operation, installation, testing, governing and gas production, measure up to the most exacting requirements, always keeping in mind the prefatory statement as to the selected scope. The various subjects are explained in simple, direct phraseology, devoid of unnecessary and bombastic word exercise; the exposition of principles is sound and thoroughly intelligible; the selection of examples from practice is judicious, and there is a conspicuous absence of that abomination, padding from builders' catalogs, which characterizes the work of so many "authors" who undertake to present engineering practice on paper. The book has a field of its own, lying between that of the very elementary type written for engine runners and that of the strictly scientific college textbook, and we heartily commend it to that field.

HANDBOOK ON THE GAS ENGINE. By Herman Häder. Translated from the German and edited by W. M. Huskisson. Published by McGraw-Hill Book Co., New York. Cloth; 330 pp., 6½x9 in.; numerous illustrations and tables. Price, $5.

This is the first comprehensive reference book relating to gas engines that has appeared in English. (The reviewer does not overlook Diederich's translation of Güldner's monumental work, but regards that as very much more than a reference book.) It is well arranged and the treatment is for the most part commendable. The definition of the word "cycle" on the first page is childishly wrong, but as the book is intended for the use of trained engineers that will do no harm. On page 2, however, is a statement which the translator should have edited heroically: "Single-acting two-[stroke-]cycle engines are seldom used and will not be considered in this book."

Chapter I is also weak in that it contains elementary comparisons with steam-engine operation which are inappropriate in a book of this class. This occurs also in other parts of the book, and there are many instances of dogmatism and narrow consideration throughout it. One of the most conspicuous of the latter kind is the presentation on page 19 of the ratio of actual to maximum possible explosions per minute, represented by the symbol €. This applies only to hit-and-miss engines and should not be put forward as a general fundamental.

An instance of dogmatism is the statement on page 125 that trunk pistons without water cooling are almost invariably adopted for engines up to 120 hp. The inference, of course, is that all engines of 125 hp. must have water-cooled pistons. There are many engines in this country developing much more than that power with uncooled trunk pistons.

Exhaustive discussions of the design of flywheels, crankshafts and pistons are given in Chapter VII, but the reviewer is unable to find any data that would be of service in working out the simplest flyball governor, to say nothing of the Hartung and Jahns types and the still more complex flywheel type.

Chapters V and VI present excellent discussions, though somewhat verbose and rather empirical in spots, of turning moments and of the forces acting on the

main bearings and the reciprocating parts. The tone is much like that of a pompous and prolix college professor delivering a lecture. This verbosity characterizes almost the entire book and mars an otherwise praiseworthy effort, besides wasting much paper and ink. Flights of scientific oratory are out of place in an engineer's reference book.

Nevertheless, this book contains a vast amount of useful data not to be found in any other English publication (always excepting Güldner), and the author may easily be forgiven his fondness for circumlocution.

MACHINE DRAWING AND SKETCHING FOR BEGINNERS. By J. H. Robson. Published by D. Van Nostrand Co., New York. Size, 5x8 in.; 191 pp.; numerous illustrations and tables. Price, $2, net.

This book is intended for beginners and engineers desirous of taking up mechanical drawing without the aid of an instructor.

The book contains a chapter briefly describing the materials used in machine construction, another on work-shop practice and appliances, and still another is devoted to questions and answers taken mostly from examination papers.

The author has endeavored to assist the student in acquiring a knowledge of machine details by providing illustrations of various objects, many of them being fully dimensioned.

The book is particularly valuable to the engineer or student, with a practical knowledge of mechanics, who is desirous of taking up the study of machine construction.

MECHANICAL WORLD YEAR BOOK FOR 1912. Published by Emmott & Co., Manchester, Eng. Boards; 285 text pages, 4x6 in.; 100 illustrations; many tables. Price, 6d.

The twenty-fifth edition of this little annual contains new sections on verniers and micrometers, gaging cylindrical bores, roller bearings, helical springs and milling-machine work. There are also several new tables and the steam-turbine section has been expanded. The sections on gas engines, oil engines and gas producers are so limited as to be of little or no value for reference purposes. There is not even a table of engine constants, and some of the statements are out of date.

What Scale Does to Boilers

The interesting and comprehensive compilation of data on the care and treatment of boilers, published by the Harrison Safety Boiler Works, Seventeenth and Clearfield Sts., Philadelphia, Penn., now appears in the fourth edition, with uptodate and amplified statistics and considerable new engineering matter, especially in regard to recent theories in connection with the corrosion of iron by boiler-feed water.

A new chart, credited to Dr. Reutlinger, of Cologne, shows that boiler plates covered with scale may rise to temperatures two or three times as great as will clean plates transmitting heat at the same rate.

The chapter on corrosion, particularly that section relating to the rôle of air in corrosion, in which the latest results of Profs. Woodbridge and Walker, of the Massachusetts Institute of Technology; Burgess, of the University of Wisconsin, and Heyn and Bauer, of Germany, are quoted, should make interesting reading for anyone having trouble of this kind. The most valuable result brought out by these investigations is the fact that corrosion may be wholly or almost entirely prevented by heating the water by spraying through an open steam bath, thereby driving off the air and other gases. Copies of this 40-page booklet will be sent free to those interested.

Steam Tables for Condenser Work

The ordinary steam tables leave much to be desired for condenser work. The Wheeler Condenser & Engineering Co., Carteret, N. J., has issued, presumably for gratuitous distribution, a little handbook of more extended steam tables for pressures below that of the atmosphere; also including a discussion of the use of the mercury column, and constants for the correction of its indications. The tables were especially calculated for this book by Prof. Marks, and are based upon the more comprehensive tables of Marks and Davis.

There are three separate tables, the first giving the properties of steam for each tenth of an inch of vacuum from 29.8 in. to 24 in. and above this by even inches to atmospheric pressure, the vacuum being referred to a 30-in. barometer. The second table gives the properties for each degree of temperature between 32 and 212 deg., the corresponding pressures being expressed in pounds per square inch, inches of mercury (32 deg.) and inches of vacuum. The third table goes by even pounds from a gage pressure of zero to 200.

It was time for somebody to do this, and it is appropriately done by a condenser company.

Exhaust Steam Engineering

"Cochrane Heaters, their Use in Steam Power Plants for Heating and Purifying Water for Boiler Feed and Other Purposes," is the title of a 144-page publication issued by the Harrison Safety Boiler Works, of Philadelphia. The topics include the choice of a system for heating boiler-feed water; relative economics of electric and direct-driven auxiliaries vs. steam-driven auxiliaries where the exhaust from the latter is used for heating the boiler-feed water; action of spraying the water through a steam bath in driving off air and gases and thereby preventing corrosion as well as bringing about the precipitation of carbonates from the water; comparison of the thermal efficiencies of direct-acting pumps, throttling plain slide-valve engines, automatic engines, compound engines and steam turbines; effect upon fuel economy of heating boiler-feed water; steaming capacity of boilers; apparatus for utilizing exhaust steam, including feed-water heaters, oil separators, traps, tanks and automatic cold-water regulators; influence of scale upon boiler-repair bills; heating of feed water in condensing plants; utilization of exhaust steam for heating and drying work, and for the operation of low-pressure turbines; operation of open feed-water heaters in multiple; pumping of hot water; limitations of injectors; a comparison of the efficiencies and other characteristics of open feed-water heaters; analysis of the design of the several parts of an open feed-water heater; analysis of the fundamental facts of water softening as it affects central stations, manufacturing plants, etc.; the metering of boiler-feed water. Besides the catalog naturally describes the several appliances manufactured by the company for use in connection with the above mentioned branches of engineering.

Copies of this publication will be sent free to owners, engineers and operators of steam plants and others having direct interest in the subjects treated.

"Mechanical Stresses in Transmission Lines," by A. Guell, has just been issued as Bulletin No. 54 of the engineering experiment station of the University of Illinois. Of the stresses to which lines of wire for the transmission of electric current are subjected, this bulletin discusses the following: (1) dead weight; (2) wind pressure; (3) change of temperature. Approximate and exact formulas are derived for use in the calculation of such stresses. An example given shows that for very long spans and great sags the parabola method of calculation, commonly used by engineers, is inadequate. Evidence is presented to show the superior advantage of the catenary method, and also that this method is sufficiently simple and accurate to serve the purpose of practice. Copies of this bulletin may be obtained gratis upon application to W. F. M. Goss, director of the engineering experiment station, University of Illinois, Urbana, Ill.

PERSONAL

A. R. Maujer, heretofore an associate editor of POWER, attached to the main office in New York City, has been appointed Western editor, with headquarters at the Chicago office.

Moments with the Ad. Editor

Here's an interesting letter from a POWER subscriber in Brownsville, Texas—interesting because it points a moral both to advertiser and reader of advertising.

It shows the advertiser that the reader nowadays wants not only to be assured that a device is good, but also "why it will pay him to use it."

It shows the reader that reading alone isn't sufficient to get the utmost value out of the selling section—

That when an advertiser has shown the way to secure greater economy the thing to do is to act—to investigate.

1. The ad. of the General Electric Co. gives me the strongest impulse to investigate their product because, in the first place, it attracts my attention by a picture which makes a distinct impression upon my mind, while the reading matter conveys to me, in a brief manner, the fact that there is an instrument which will give correctly the amount of work each boiler is doing, a vital point in every steam plant.

Further, it mentions how, by having this information, many losses may be prevented and the general efficiency of the boiler improved.

In the second place it assures me that this instrument is not complicated or costly, as I might otherwise suppose was the case with an instrument of this kind.

Lastly, it requests me to write for a bulletin, describing the instrument more fully.

2. The steam trap ad. of the Geo. M. Newhall Co. seems to me to have the biggest educational value because it gives, both by illustration and description, a clear idea of the construction of this device and calls special attention to six of its strong points, and explains why these points are strong ones.

It also has a further educative value by telling where a person can secure, free of charge, a booklet fully describing this trap and steam traps in general.

3. The ad. of the General Electric Co. appeals to me as being the best all round ad. for the following reasons—first, it has a distinct, life-like picture to attract a person's attention. Second, it sets forth a fact in the first paragraph which holds the interest of the reader and compels him to read further, for every first-class engineer is interested in the amount of work each particular boiler in his plant is doing.

Third, the ad. is printed in large clear type, is to the point, and is so arranged that the reader reads it through in a straightforward way, and has no inclination to pass it by, until he has read it all.

Fourth, it gives just enough information to interest the engineer to the extent that he wants to know more of this particular instrument, and hence is ready to accept the invitation to send for a bulletin.

4. I have answered the ad. of the General Electric Co.

Another letter which furnishes food for thought for both advertiser and reader is this one from a subscriber in Fort Morgan, Ala.:

1. The Lytton trap "ad." They mention the fact that their trap is installed in the Delray station in Detroit, under the severest of conditions, and is giving satisfaction. This simple statement is more convincing than a half dozen long drawn out reasons why their trap is superior to others. In reading descriptions of modern plants I always notice the *kind* of apparatus installed, as there always is a reason.

2. Smooth-On "ad." Each week since I have been reading POWER I notice where some engineer has been helped over a hard place by using Smooth-On. When the water-jacket on our gas engine cracked during the recent cold spell, it was perfectly natural for me to say, "Smooth-On will fix it."

3. Ambest lubricated metallic packing "ad." The picture has a distinct educational value; it shows the construction and character of the packing; it shows how easy it is to pack a rod, and the method of packing. The arguments are logical and easily understood. The offer of a free testing sample should bring the advertiser big results.

4. I answered the Lytton and Rochester "ads."

It is true that it will only pay that manufacturer to advertise whose goods will help you improve conditions in your plant, will help you secure economical results—

But remember that this selling section will help *you* best only if you take advantage of the investigation invitations which the advertisers extend to you.

Old Junk in Steam Main

By W. F. Fischer

Jim, the steam fitter, was called into a comparatively new plant one Sunday morning to assist the engineer in making some repairs to the steam piping, which appeared to be obstructed. The engineer had been experiencing trouble for some time with the valves in the main steam header, one of which he could not close. As the plant was shut down over Sunday and as there was only one spare valve available, this time was chosen to make the repairs.

Upon removing the first valve they found an old piece of cloth firmly wound around the valve stem and otherwise tangled up with the interior mechanism of the valve. Upon removing this they discovered a brass button bearing the trademark of a well known manufacturer of overalls.

The valve stem was straightened, the seat reground, and the valve replaced in the line. Upon taking down the second valve they found, as Jim expressed it, sticking in the end of the line just at the entrance a badly frazzled bunch of stewed leather which proved to be the remains of an old shoe, part of which had been torn away and was wedged firmly between the seat and the disk of the valve.

Jim and the engineer then got their heads together and began to reason as follows: A shoe in the hand means one more in the steam main, and there should be at least one more shoe and a pair of overalls stowed away some place in the piping, provided they had not been chewed up by an old engine which the engineer remarked was a great "steam eater."

The next question was how to locate the junk without taking down the whole steam header and perhaps several of the branch mains. While they were debating, Jim spied a steam separator at the end of the line, with a side inlet and top outlet, and as this seemed to be as good a resting-place as any for old junk, they uncoupled the separator and found the other old shoe and the overalls.

Further investigation disclosed a bolt about 3½ in. long, three hexagon nuts and a number of small pieces of iron. The separator was cleaned and replaced in the line, the new valve was installed and the plant was ready for operation the first thing in the morning.

The old shoes and overalls were probably hid away in the piping during erecting by some careless steam fitter who afterward forgot them.

NEW EQUIPMENT

ATLANTIC COAST STATES

Winooski, Vt., is contemplating the installation of a municipal electric-light plant.

Plans and specifications are being prepared for additional equipment in the water-works plant at North Attleboro, Mass. William Plattner is engineer.

The Long Island R.R. will construct a new power house and substation at Cedarhurst, N. Y. G. B. Gunn, Jamaica, N. Y., is superintendent of construction.

The Jamestown Lighting & Power Co. will construct an electric-light plant at Celoron, N. Y. Frank W. Bullock, Jamestown, N. Y., is superintendent.

The Western New York Water Co., Buffalo, N. Y., will install new electrically driven pumps in the new pumping station which it is constructing at Niagara Falls.

Two gas engines will be installed in the office building at 141 Fifth Ave., New York, N. Y. They will be used for generating light. Charles Rody is chief engineer.

Plans are being prepared by the Rochester Ry. & Light Co., Rochester, N. Y., for the construction of a new power plant. J. T. Hutchings is general manager.

The Burden Iron Co., Troy, N. Y., will soon install a generating station to furnish current for motors, and arc and incandescent lighting. Plans are being prepared by Andrew Kidd, Jr., 92 Liberty St., New York.

Burlington, N. J., is considering the installation of an electric-light plant in connection with the water-works system. Estimated cost, $18,000.

E. J. Brooks & Co., Newark, N. J., will increase the capacity of their power plant. A new engine will be installed.

C. B. Seedenberg and H. C. Hartman will install an electric-light plant at Martinsburg, Penn.

The Pittsburg Rys. Co., Pittsburg, Penn., will expend about $699,102 for power houses and substations.

The Johnstown & Altoona Ry. Co., Johnstown, Penn., will build a new power plant. Plans are being prepared by Pierce & Barnes, 7 Water St., Boston, Mass.

George H. Bowman and J. E. Newcomer, of Hagerstown, Md., contemplate the construction of an electric-power plant on the Antietam Creek, near Leitersburg, Md.

Homer, N. Y., contemplates the installation of a municipal electric-light plant.

The Lewiston & Lake Ontario Shore Power Co., Niagara Falls, N. Y., is considering the construction of a transformer station at Lewiston, N. Y.

A new water-works system will be constructed at Mount Morris, N. Y. J. A. Striker is village clerk.

Chester & Fleming, Union Bank Bldg., Pittsburg, Penn., have prepared plans for the construction of a filtration plant at Phoenixville, Penn.

Trenton, N. J., has decided to build a 30,000,000-gal. filtration plant. A. C. Gregory is engineer of sewers and water supply.

SOUTHERN STATES

The Highland Park Manufacturing Co., Charlotte, N. C., will install a 250-kw., 2300-volt, three-phase, 60-cycle generator.

The Southern Power Co. will construct a substation at Mount Holly, N. C. Estimated cost, $100,000. C. I. Burkholder, Charlotte, is manager.

The R. T. Reynolds Tobacco Co., Winston Salem, N. C., will soon install a 1000-hp. central power station.

Barnwell, S. C., contemplates the installation of an electric-light plant and water-works system.

The Deep Creek Lumber Co., Norfolk, Va., contemplates the installation of an electric-light and power plant near Lake City, S. C.

The council, Florence, S. C., contemplates a bond issue of $35,000 for the installation of an electric-light plant. W. J. Wilkins is chairman of board of public works.

The Savannah Electric Co., Savannah Ga., is considering the construction of a new power plant, to have a capacity of 40,000 kw., and to cost approximately $1,000,000. M. L. Sperry is manager.

Bids will be received by George M. Robbins, chairman of trustees of city bonds, Titusville, Fla., until noon, May 20, for the construction of a water-pumping station.

The Schantz Electric, Ice & Water Co. will construct a water-works and ice plant at Daytona Beach, Fla. E. L. Gunn is general manager.

The finance committee of the city council of Paris, Ky., is receiving estimates as to the cost of erecting a municipal electric-light plant.

Cheraw, S. C., is contemplating the construction of a water-works system. Estimated cost, $47,000.

Dothan, Ala., will soon award contract for the installation of an electric street-lighting system.

CENTRAL STATES

Granville, Ohio, is considering the installation of an electric-light plant.

The Dayton, Covington & Piqua Traction Co. will install a new 150-kw. generator and two 150-hp. boilers in its substation at West Milton, Ohio. Thomas J. Brennan is general superintendent.

Power-plant equipment will be required for the four-story warehouse to be erected for J. Neal at Seventy-second St. and Euclid Ave., Cleveland, Ohio. Knox & Elliott have completed the plans.

The Auglaize Power Co. will build a water-power electric plant on the Auglaize River, near Delphos, Ohio. Estimated cost, $500,000. James S. Ashley is president.

Mansfield, Ohio, is considering the installation of a municipal electric-light plant and a street-lighting system. Estimated cost, $132,770. E. A. Merkel is city engineer.

J. E. Pickens, Kankakee, Ill., is preparing plans for a water-works system at Chebanse, Ill. Estimated cost, $20,000.

The Business Men's Association of Almond, Wis., is contemplating the installation of a municipal electric-light plant.

Plans are being prepared by W. G. Kirchoffer, Madison, Wis., for a water-works system at Horicon, Wis. Estimated cost, $45,000.

Benton Harbor, Mich., has plans for a 5,000,000-gal. pumping station and a water-works plant.

Jackson, Mich., will install two 125-hp. boilers. A. W. D. Hall is city engineer.

Crooksville, Ohio, will install a water-works system. Plans are being pre-

pared by H. L. Maddox, Newark Trust Bldg., Newark, Ohio.

The Scioto Sand Co., Portsmouth, Ohio, will install new power-plant equipment in its plant.

Manchester, Mich., will install a water-works system at a cost of $27,000.

WEST OF THE MISSISSIPPI

Covington, Iowa, is considering the installation of a municipal electric-light plant. G. E. Phillips is mayor.

S. D. Henry contemplates the installation of an electric-light and power plant, at Coon Rapids, Iowa.

The Des Moines City Ry. Co., Des Moines, Iowa, contemplates the expenditure of $95,584 for new power-plant equipment. J. E. Ralston is superintendent of power.

The Fort Dodge Light & Power Co. is considering the installation of a steam-heating plant in connection with its electric-light plant, at Fort Dodge, Iowa. Richard M. Parker is general manager.

Pomeroy, Iowa, contemplates the installation of an electric-light plant.

Paullina, Iowa, is considering the construction of a pumping station. Estimated cost, $8000.

Rochester, Minn., is considering plans for improvements in the municipal electric-light plant. J. A. Eichinger is superintendent of plant.

Bids will be received by the state board of control, Topeka, Kan., until May 7, for the installation of a 30-kw. direct-current generator attached to a 50-hp. side-crank high-speed engine for the school for the deaf, Olathe, Kan. Charles W. Gibbs is secretary of board.

Lyndon, Kan., will install a water-works system. Estimated cost, $39,000. L. F. Hussey is mayor.

Rosalie, Neb., has voted bonds for $8000 for the construction of a water-works system. Grant & Letton, Lincoln, Neb., are engineers.

Nisland, S. D., is considering the installation of a municipal electric-light plant and water-works system.

The installation of an electric-light plant, at Nome, N. D., is contemplated. C. E. Greenwood is interested.

C. C. Cunningham will construct an electric-light and power plant at Baggs, Wyo.

Cody, Wyo., is considering the installation of a filtration plant and water-works system, at a cost of $65,000. Burns & McDonnell, Scarritt Bldg., Kansas City, Mo., are consulting engineers.

The municipal light and power plant, at Butler, Mo., will install additional generating equipment. Charles W. Miller is engineer of power station.

The Dorris Motor Car Co., St. Louis, Mo., will build a power plant, at Sarah St. and Laclede Ave. Estimated cost, $7200.

Gravette, Ark., is preparing plans for the construction of a water-works system and electric-light plant. Estimated cost, $25,000.

The Camp Hardy Corporation, Hardy, Ark., is considering the installation of an electric-light plant, in connection with its establishment.

The Cuero Light & Power Co., Cuero, Tex., is planning improvements to its plant. E. J. Davis is manager.

The Northern Texas Traction Co. will install an additional 5000-kw. turbine in its power station at Handley, Tex. George H. Clifford, Fort Worth, Tex., is vice-president and manager.

Grandview, Wash., will install a water-works system. Estimated cost, $18,000. A. E. Fish is town clerk.

The city of Carlton, Ore., will install a water-works system. Estimated cost, $34,000. U. Kutch is city recorder.

Monmouth, Ore., will install a water-works system, at a cost of $20,000. J. W. White is a member of the water committee.

G. G. Bushman contemplates the installation of an electric-light plant at Florence, Ore.

The substation at Isleton, Calif., of the Great Western Power Co., was recently destroyed by fire, causing a loss of $10,000.

The Pacific Gas & Electric Co. is considering plans for remodeling its plant at Wheatland, Calif. John A. Britton, 445 Sutter St., San Francisco, is general manager.

The Universal Electric & Gas Co. contemplates the construction of a power plant at First and Barannon Sts., San Francisco, Calif.

The Northern California Power Co. is considering the installation of new equipment in its substation at Chico, Calif. E. V. D. Johnson, Redding, is Mgr.

Ely, Iowa, is considering plans for the installation of an electric-light plant.

William Richard contemplates the construction of an electric-light plant, at Shell Rock, Iowa.

The installation of a municipal electric-light plant, at Hampton, Iowa, is contemplated.

Laurens, Iowa, is considering the construction of a municipal electric-light plant.

Burr Oak, Kan., will install a water-works system, to cost $25,000.

Lennox, S. D., contemplates the installation of an electric-light plant.

The council, Northwood, N. D., is considering the installation of an electric-light plant.

Charles F. Hoff is considering the installation of a water-works plant, at Rockport, Tex.

Ventura, Calif., is contemplating the installation of a municipal electric-light plant.

Modesto, Calif., has voted a bond issue of $82,500 for the construction of a water-works system.

The London & Northwestern Ry. Co. is preparing to construct a power plant, at Rock Glen, near Arkona, Ont. Smith, Kerry & Chace, Toronto, are engineers-in-charge.

Kindersley, Sask., is taking steps for the installation of a municipal electric-light and power plant. Estimated cost, between $10,000 and $15,000.

The New England Fish Co., Seattle, Wash., will build a new cold-storage plant, at Ketchikan, Alaska. Estimated cost, $73,000.

Friday Harbor, Wash., will receive bids until Apr. 13, for constructing a water-works system. V. J. Whitney, Globe Bldg., Seattle, is engineer. Estimated cost, $15,000.

The Dealers Ice & Cold Storage Co., of Globe, Ariz., will install an ice and cold-storage plant, at Miami, Ariz. The ice plant will have a daily capacity of 15 tons.

BUSINESS ITEMS

C. Lee Straub, consulting engineer, of New York City, formerly with the engine department of the Power & Mining Machinery Co., of Milwaukee, Wis., has been placed in charge of the lubricating department of the Richardson-Phenix Co., Hudson Terminal Building, New York City.

ENGINEERS WANTED

Under this head are published, without charge, advertisements of actual vacancies in power plants for operating engineers and assistant engineers. For the guidance of applicants for advertised positions, the advertisements must contain particulars to show the class of service demanded and the wages paid. Employers who desire to advertise under this head are requested to write for a blank form of application.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

SELLING AGENTS wanted in important centers for Betson plastic fire brick; see page 142 in this issue.

SALESMAN — Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," Power.

AGENTS for first-class steam specialty in use throughout United States. Address C. S. Wood, 410 S. 15th St., Philadelphia, Penn.

THE VULCAN SOOT CLEANER offers an exceptional opportunity for power specialty salesman Address G. L. Simonds & Co., 802 Steinway Bldg., Chicago, Ill.

AGENTS WANTED to sell forced draft blowers and damper regulators; will give exclusive territory to right parties; commission on sales. Turbo-Blower Co., 30 Church St., New York.

WANTED—Thoroughly competent experienced mechanical rubber salesman for the city of Chicago; must have an acquaintance among the large buying trade; state age and experience; also give references. Address Box 814, Trenton, N. J.

AGENTS selling to steam plants can make big money selling for us; no samples; no experience is necessary. Armstrong Mfg. Co., Springfield, Ohio.

REPRESENTATIVES WANTED in every territory to sell high grade staple product on profit sharing basis to power plants of all description; excellent side line. Full particulars on application to P. O. Box 404, Newark, N. J.

CHIEF ENGINEER for central station and railway steam plant, northern Michigan; we are now remodeling the plant and want a man to keep things up and get efficiency; state experience in full and salary expected. Box 617, Power.

A YOUNG MECHANICAL engineer familiar with the operation of power plants, able to carry out tests of boilers, gas producers, gas engines and be familiar with all kinds of measuring instruments; applicant must give full information concerning his training and experience. Box 615, Power.

MANUFACTURERS' AGENTS to handle thoroughly guaranteed steam specialties: feed water heaters, steam and oil separators, traps and exhaust pipe heads; not necessary to represent entire line if interested in only part of it: our specialties are well advertised and of good reputation. Box 569, Power.

SITUATIONS WANTED

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

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PATENTS—Herbert Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make a free search and report if patent can be had and the exact cost.

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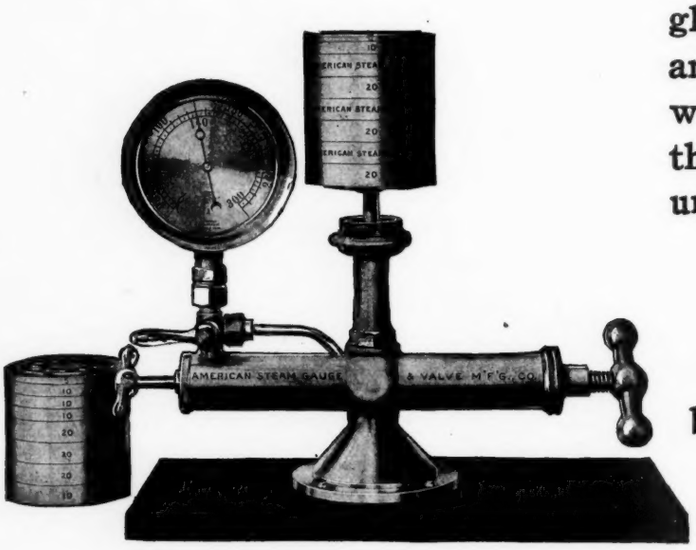

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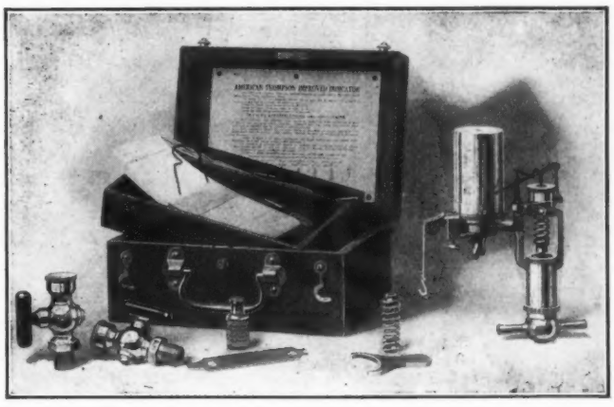

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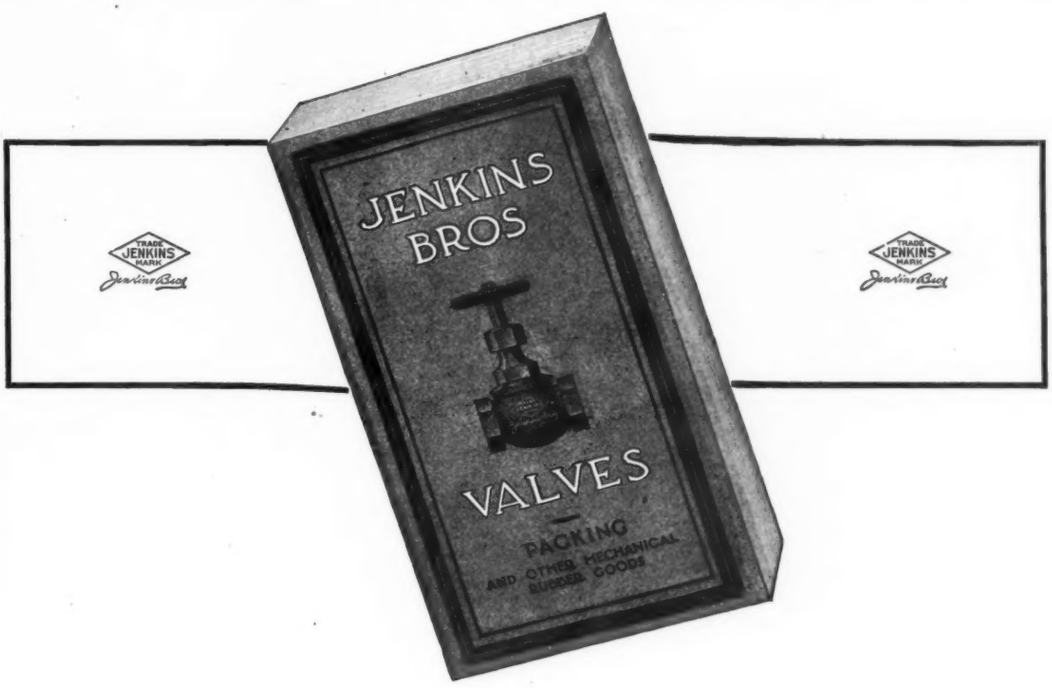

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